MATERIALS ENGINEERING

MATERIALS SCIENCE VS. MATERIALS ENGINEERING??

MATERIALS SCIENCE

INVOLVES **INVESTIGATING** THE RELATIONSHIPS THAT EXIST BETWEEN THE STRUCTURES AND PROPERTIES OF MATERIALS.

MATERIALS ENGINEERING

IS ON THE BASIS OF THE STRUCTURE-PROPERTY CORRELATIONS, **DESIGNING** OR ENGINEERING THE STRUCTURE OF A MATERIAL TO PRODUCE A PREDETERMINED SET OF PROPERTIES.

THE PERIODIC TABLE

IN SEVEN HORIZONTAL ROWS

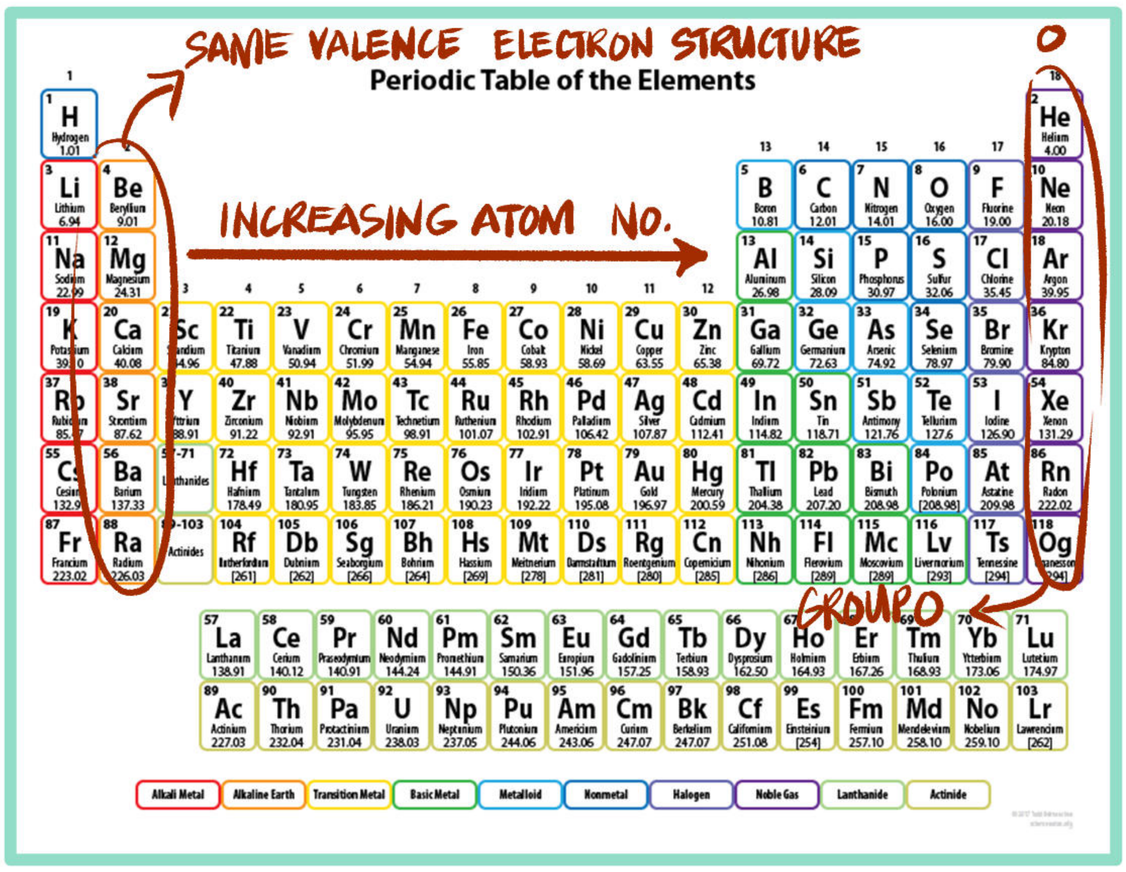
INCREASING ATOMIC NUMBER FROM LEFT TO RIGHT

IN ALL COLUMNS

THE SAME/SIMILAR VALENCE ELECTRON STRUCTURES

GROUP 0 (RIGHT MOST GROUP)

ARE INERT GASES, WHICH HAVE COMPLETELY FILLED ELECTRON SHELLS AND STABLE ELECTRON CONFIGURATIONS. ALL OF THEM ARE CHEMICALLY NON-REACTIVE.



6 CLASSIFICATION OF MATERIALS

METALS	USED IN STRUCTURAL APPLICATIONS.
CERAMICS	ARE COMPOUNDS BETWEEN METALLIC & NONMETALLIC ELEMENTS.
COMPOSITES	CONSISTS OF COMBINATIONS OF TWO OR MORE DIFFERENT MATERIALS.
POLYMERS	INCLUDE PLASTIC AND RUBBER MATERIALS
SEMICONDUCTORS	ARE INTERMEDIATE BETWEEN THE ELECTRICAL; CONDUCTORS AND INSULATORS.
BIOMATERIALS	ARE EMPLOYED IN COMPONENTS IMPLANTING INTO THE HUMAN BODY FOR REPLACEMENT.

ATOMIC BONDING IN SOLIDS

IONIC BONDING

- ATOMS OF A **METALLIC ELEMENT** EASILY GIVE UP THEIR VALENCE ELECTRONS TO THE **NONMETALLIC ATOMS**. (METAL AND NON-METAL)

COVALENT

- BY THE **SHARING OF ELECTRONS** BETWEEN ADJACENT ATOMS

METTALIC

- THE VALENCE ELECTRONS FORM A **"SEA OF ELECTRONS"** THAT IS UNIFORMLY DISPERSED AROUND THE METAL ION CORES AND **ACTS AS A FORM OF GLUE FOR THEM**
- GOOD CONDUCTORS!!

PROPERTIES OF MATERIALS

MECHANICAL PROPERTY

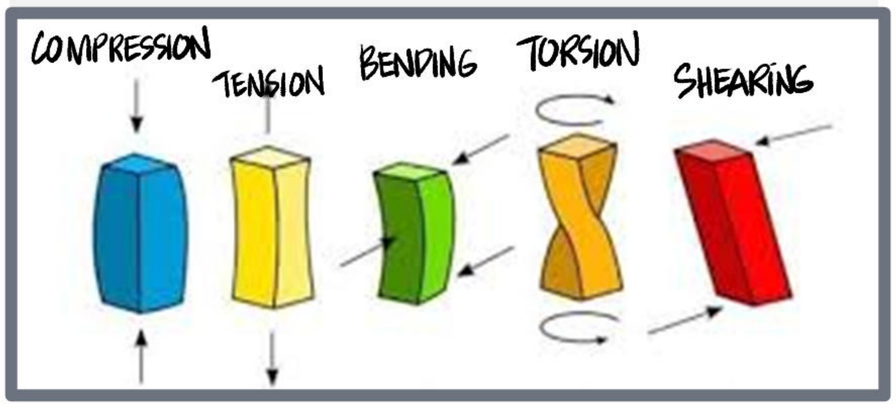
- IS A **MEASURE OF A MATERIAL'S ABILITY TO CARRY OR RESIST MECHANICAL FORCES OR STRESSES**. IT IS USUALLY DETERMINED BY SUBJECTING PREPARED SPECIMENS TO STANDARD LABORATORY TESTS. IT RELATES TO ON HOW A MATERIAL RESPOND TO APPLIED LOADS (OR FORCES).

STATIC MECHANICAL PROPERTIES

STRENGTH

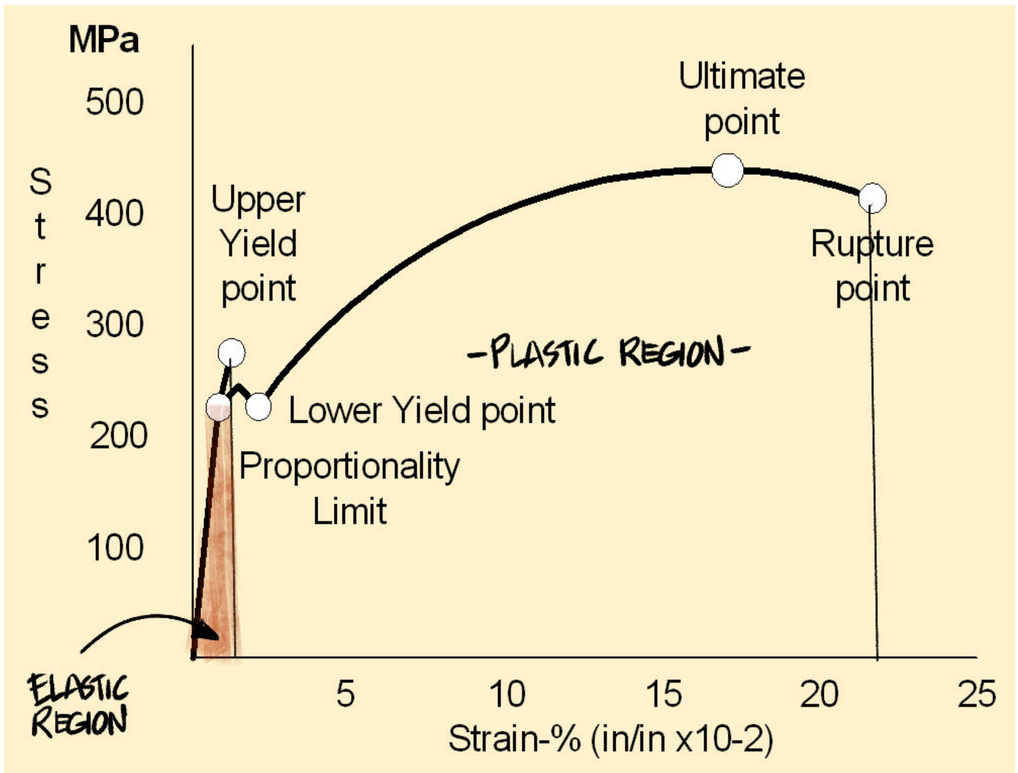
- IT IS THE ABILITY OF A MATERIAL TO **RESIST THE EXTERNALLY APPLIED** FORCES WITHOUT BREAKING OR YIELDING.
- THE INTERNAL RESISTANCE OFFERED BY A PART TO AN EXTERNALLY APPLIED FORCE IS CALLED **STRESS**.

TYPES OF STRESSES



$$\text{STRESS}(\sigma) = \frac{\text{FORCE}}{\text{AREA}} = \frac{F}{A}$$

ENGINEERING STRESS-STRAIN CURVE



YIELD POINT	IS THE STRESS VALUE REACHED WHERE ADDITIONAL STRAIN OCCURS WITHOUT FURTHER IN STRESS.
UPPER YIELD POINT	IS THE HIGHEST STRESS PRECEDING EXTENSIVE STRAIN.
LOWER YIELD POINT	IS THE LOWER, RELATIVELY CONSTANT, RUN-OUT VALUE. IT IS THE ONE THAT USUALLY APPEARS IN THE TABULATED DATA
OFFSET YIELD STRENGTH	IS THE VALUE OF THE STRESS THAT WILL PRODUCE A GIVEN, BUT TOLERABLE, AMOUNT OF PERMANENT STRAIN.
ULTIMATE TENSILE STRENGTH	(OFTEN CALLED TENSILE STRENGTH) IS THE MAXIMUM STRESS, IN TENSION, THAT MAY BE SUSTAINED WITHOUT FRACTURE.
FRACTURE STRENGTH	IS THE STRESS AT WHICH FRACTURE OCCURS.

STRAIN

- IS THE UNIT DEFORMATION OF AN OBJECT DUE THE APPLIED LOAD.

$$\varepsilon = \frac{\Delta L}{L_o}$$ | ΔL = ELONGATION, L_o = ORIGINAL LENGTH

STIFFNESS

- IT IS THE ABILITY OF A MATERIAL TO **RESIST DEFORMATION UNDER STRESS.**
- **THE MODULUS ELASTICITY IS THE MEASURE OF STIFFNESS.**

ELASTICITY

- IT IS THE PROPERTY OF A MATERIAL TO **REGAIN ITS ORIGINAL SHAPE** AFTER DEFORMATION WHEN THE EXTERNAL FORCES ARE REMOVED.
- FOR SOME MATERIALS THE **ELASTIC LIMIT AND PROPORTIONALITY LIMIT** ARE ALMOST IDENTICAL. *IN MOST CASES, HOWEVER THE ELASTIC LIMIT IS SLIGHTLY HIGHER.*

PLASTICITY

- IT IS PROPERTY OF A MATERIAL WHICH **RETAINS THE DEFORMATION PRODUCED** UNDER LOAD PERMANENTLY.
- THIS PROPERTY OF THE MATERIAL IS NECESSARY FOR FORGINGS, IN STAMPING IMAGES ON COINS AND IN ORNAMENTAL WORK.

PLASTICITY VS. ELASTICITY??

COIN

SPRING

PROPORTIONALITY LIMIT

- IS A POINT ON A STRESS-STRAIN CURVE AT WHICH THE **STRAIGHT LINE PROPORTIONALITY BETWEEN STRESS AND STRAIN CEASES.**
- UP TO THAT POINT THE MATERIAL **OBEYS THE HOOKE'S LAW**, WHICH STATES THAT THE STRAIN IS DIRECTLY PROPORTIONAL TO THE STRESS.

$$\sigma = E\varepsilon$$ | E= MODULUS OF ELASTICITY

POISSON'S RATIO (v)

- IS THE RATIO OF THE **LATERAL STRAIN (CONTRACTION) TO THE LONGITUDINAL STRAIN (EXTENSION)** WHEN THE ELEMENT IS LOADED WITH A LONGITUDINAL TENSILE FORCE.

$$v = -\frac{\varepsilon_x}{\varepsilon_y} = -\frac{\varepsilon_y}{\varepsilon_z}$$

$$E = 2G(1+v)$$ | E = MODULUS OF ELASTICITY, G = SHEAR MODULUS

IN MOST METALS ; $G = 0.4E$

POISSON'S RATIO FOR ISOTROPIC MATERIALS	**0.25**
MAXIMUM VALUE OF POISSON'S RATIO	**0.50**
METALS AND OTHER ALLOYS, VALUES RANGES	**0.25-0.35**

BRITTLENESS

- IS ONE THAT LACKS SIGNIFICANT DUCTILITY. IT IS THE TENDENCY OF A MATERIAL TO FRACTURE WITHOUT APPRECIABLE DEFORMATION.

DAMPING CAPACITY

- IS THE ABILITY OF A MATERIAL TO **ABSORB OR DAMP VIBRATIONS**, WHICH IS A PROCESS OF ABSORBING KINETIC ENERGY OF VIBRATION OWING TO HYSTERESIS.

NOTE: CAST IRON IS A MUCH BETTER DAMPING MATERIAL THAN STEEL.

EMBRITTLEMENT

- INVOLVES THE LOSS OF DUCTILITY BECAUSE OF A PHYSICAL OR CHEMICAL CHANGE OF THE MATERIAL

MACHINABILITY

- REFERS TO THE RELATIVE EASE WITH WHICH A MATERIAL CAN BE CUT.

DUCTILITY

- IT IS THE PROPERTY OF A MATERIAL ENABLING IT TO BE **DRAWN INTO WIRE** WITH THE APPLICATION OF A TENSILE FORCE.
- IS A MEASURE OF THE DEGREE OF PLASTIC DEFORMATION THAT HAS BEEN SUSTAINED AT FRACTURE.
- IT IS THE ABILITY OF THE MATERIAL TO **CHANGE SHAPE WITHOUT FRACTURE.** IT IS REPORTED IN TERMS OF THE PERCENT ELONGATION OF A SPECIFIC GAGE LENGTH.

PERCENT ELONGATION

$$\%E = \frac{L_f - L_o}{L_o} \times 100\%$$

PERCENT REDUCTION IN AREA

$$\%RA = \frac{A_o - A_f}{A_o} \times 100\%$$

MALLEABILITY

- IS THE ABILITY OF A MATERIAL TO **BE SEVERELY DEFORMED PLASTICALLY IN COMPRESSION WITHOUT FRACTURE**
- IT IS A SPECIAL CASE OF DUCTILITY WHICH PERMITS MATERIALS TO BE **ROLLED OR HAMMERED INTO THIN SHEETS.**
- **A MALLEABLE MATERIAL SHOULD BE PLASTIC BUT IT IS NOT ESSENTIAL TO BE SO STRONG.**

ORDER OF DUCTILITY		ORDER MALLEABILITY	
C	COPPER	C	COPPER
I	IRON	A	ALUMINUM
A	ALUMINUM	T	TIN
Z	ZINC	L	LEAD
T	TIN	I	IRON
L	LEAD	Z	ZINC

RESILIENCE

- IS THE CAPACITY OF A MATERIAL TO **ABSORBED ENERGY WHEN IT IS ELASTICALLY DEFORMED** AND THEN UPON UNLOADING, TO HAVE THIS ENERGY RECOVERED.
- THIS PROPERTY IS ESSENTIAL FOR SPRING MATERIALS.

MODULUS OF RESILIENCE IS THE AMOUNT OF ENERGY THAT A UNIT VOLUME OF MATERIAL CAN ABSORB WHILE ON THE ELASTIC RANGE.

$$U_R = \int_0^{\varepsilon_y} \sigma \, d\varepsilon$$

ASSUMING A LINEAR ELASTIC REGION

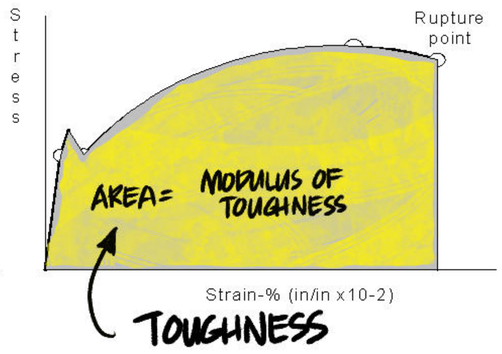
$$U_R = \frac{1}{2}\sigma_y \varepsilon$$

TOUGHNESS

- IS THE CAPACITY OF MATERIAL TO **WITHSTAND A SHOCK LOAD WITHOUT BREAKING.**
- IT IS THE ABILITY OF A MATERIAL TO **ABSORB ENERGY DURING PLASTIC DEFORMATION** UP TO RUPTURE.

MODULUS OF TOUGHNESS, T IS EQUAL TO THE TOTAL AREA UNDER THE STRESS-STRAIN CURVE TO THE POINT OF RUPTURE.

RESILIENCE VS. TOUGHNESS???



TRUE STRESS – TRUE STRAIN

TRUE STRESS

INSTANTANEOUS **APPLIED LOAD** DIVIDED BY THE INSTANTANEOUS **CROSS-SECTIONAL AREA**

$$\sigma_T = \frac{F}{A_i}$$

TRUE STRAIN

NATURAL LOGARITHM OF THE RATIO OF **INSTANTANEOUS GAGE LENGTH** TO THE ORIGINAL GAGE LENGTH

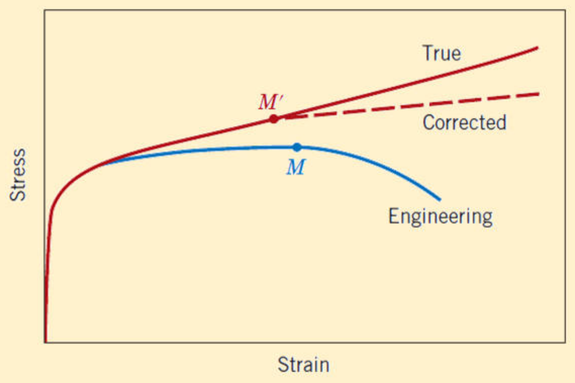
$$\varepsilon_T = \ln\left(\frac{l_i}{l_o}\right)$$

IF NO VOLUME CHANGE OCCUR DURING DEFORMATION

$$A_i l_i = A_o l_o$$

$$\sigma_T = \sigma(1+\varepsilon)$$

$$\varepsilon_T = \ln(1+\varepsilon)$$



NECKING BEGINS AT POINT M ON THE ENGINEERING CURVE, WHICH CORRESPONDS TO M' ON THE TRUE CURVE. **THE "CORRECTED" TRUE STRESS-STRAIN CURVE TAKES INTO ACCOUNT THE COMPLEX STRESS STATE WITHIN THE NECK REGION**

HARDNESS

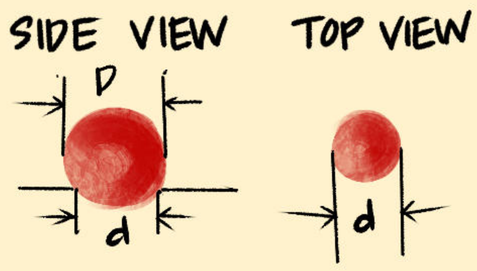
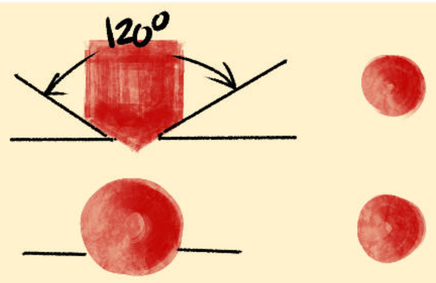
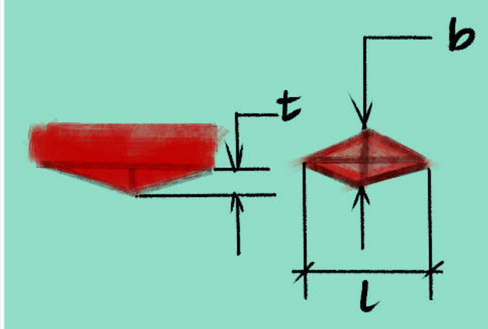
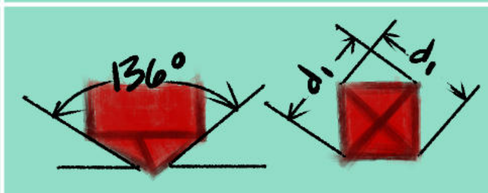
- IS A MEASURE OF MATERIAL'S **RESISTANCE TO PERMANENT DEFORMATION (INDENTATION)** UNDER STATIC OR DYNAMIC LOADING

SHOP DEFINITION: RESISTANCE TO FILING
ENG'G DEFINITION: RESISTANCE TO PENETRATION

FOR MOST STEELS:

$$TS(MPa) = 3.45 \times BHN$$
$$TS(psi) = 500\,BHN$$

TYPES OF HARDNESS TESTS

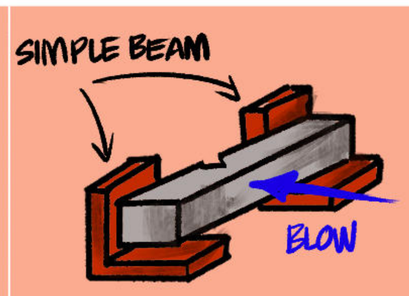
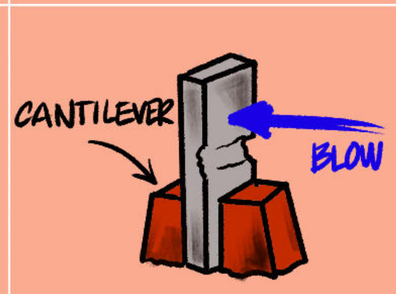
TEST	DESCRIPTION	INDENTER	SHAPE	FORMULA
BRINELL	USES **5** MM AND **10** MM DIAMETER HARDENED STEEL/TUNGSTEN CARBIDE BALLS	**10** MM SPHERE OF STEEL OR TUNGSTEN CARBINE		$HB = \frac{2P}{\pi D\left(D - \sqrt{D^2 - d^2}\right)}$
ROCKWELL	USES SMALL-DIAMETER STEEL BALL (DIAMETER= **1/16** IN. - **½ 2** IN) OR **A DIAMOND TIPPED CONE CALLED "B R A L E ".**	DIAMOND CONE STEEL BALL (DIAMETER= **1/16** IN - ½ IN)		60 100 150 } ROCKWELL 15 30 45 } SUPERFICIAL ROCKWELL
MICROHARDNESS TESTER	- IS USED WHERE IT IS NECESSARY TO DETERMINE THE **HARDNESS OF A VERY PRECISE AREA OF MATERIAL**, OR WHERE THE MATERIAL OR **SURFACE LAYER IS EXCEPTIONALLY THIN.** - LOAD APPLICATION RANGES FROM **25** TO **3600 GRAMS**			
DUROMETER	- FOR TESTING **SOFT, ELASTIC MATERIALS, SUCH AS RUBBERS AND NON-RIGID PLASTICS.** - IT MEASURES THE RESISTANCE OF A MATERIAL TO ELASTIC PENETRATION. - **SPRING LOADED CONICAL STEEL INDENTER**			
KNOOP TEST	ITS LENGTH OF INDENTATION IS MEASURED WITH **THE AID OF A MICROSCOPE.** IT IS ALSO USED TO DETERMINE THE HARDNESS OF MATERIALS WHICH ARE **EXCEPTIONALLY THIN.**	DIAMOND PYRAMID		$l/b = 7.11$ $b/t = 4.0$ $HK = \frac{14.2P}{l^2}$
VICKERS	USES A SQUARE-BASED DIAMOND PYRAMID AS AN INDENTER.	DIAMOND PYRAMID		$HV = \frac{1.854P}{d_1^2}$
UCI (ULTRASONIC CONTACT IMPEDANCE)	**AN EXTENSION OF THE LOW-LOAD RANGE VIKERS TEST** TO PERMIT RAPID, ON SITE HARDNESS DETERMINATIONS **WITHOUT THE NEED FOR MEASURING THE SIZE OF INDENTATION.**			
SCELEROSCOPE	THE HARDNESS OF A MATERIAL IS MEASURED BY THE **REBOUND OF A SMALL DIAMOND-TIPPED HAMMER** THAT IS DROPPED FROM A FIXED HEIGHT ONTO THE SURFACE OF THE MATERIAL TO BE TESTED. **IT MEASURES THE RESILIENCE OF A MATERIAL**			
FILE TEST	MEASURES THE RESISTANCE OF A MATERIAL AGAINST SCRATCHING			

DYNAMIC MECHANICAL PROPERTIES

- SUDDEN LOADS (IMPACTS) OR LOADS THAT VARY RAPIDLY IN MAGNITUDE
- REPEATED CYCLES OF LOADING AND UNLOADING
- FREQUENT CHANGES IN THE MODE OF LOADING SUCH AS FROM TENSION TO COMPRESSION.

IMPACT TESTS

- IS A TEST USED **TO MEASURE THE ENERGY ABSORBED DURING THE FRACTURE OF A SPECIMEN** OF STANDARD DIMENSIONS AND GEOMETRY WHEN SUBJECTED TO VERY RAPID (IMPACT) LOADING.

CHARPY TESTS (BENDING IMPACT)	SUPPORTED AT **BOTH ENDS** BY SIMPLE BEAM	
IZOD TESTS (BENDING IMPACT)	SUPPORTED BY CANTILEVER BEAM	
TENSION IMPACT		

METAL FATIGUE

- IS A PHENOMENON WHICH MAY RESULT FROM THE **CYCLIC REPETITION OF A PARTICULAR LOADING CYCLE** OR FROM ENTIRELY RANDOM VARIATIONS IN STRESS. IT ACCOUNT FOR NEARLY **90%** OF ALL MECHANICAL FRACTURES.

ENDURANCE LIMIT

- IS THE LIMITING (MAXIMUM) **STRESS** BELOW **WHICH THE MATERIAL CAN PRESUMABLY ENDURE AN INFINITE NUMBER OF LOAD CYCLES.**

CREEP

- IS A **SLOW PROCESS** OF PLASTIC DEFORMATION THAT TAKES PLACE WHEN A MATERIAL IS SUBJECTED TO A **CONSTANT CONDITION OF LOADING BELOW ITS NORMAL YIELD STRENGTH.**

DESIGN/SAFETY FACTORS

DESIGN STRESS

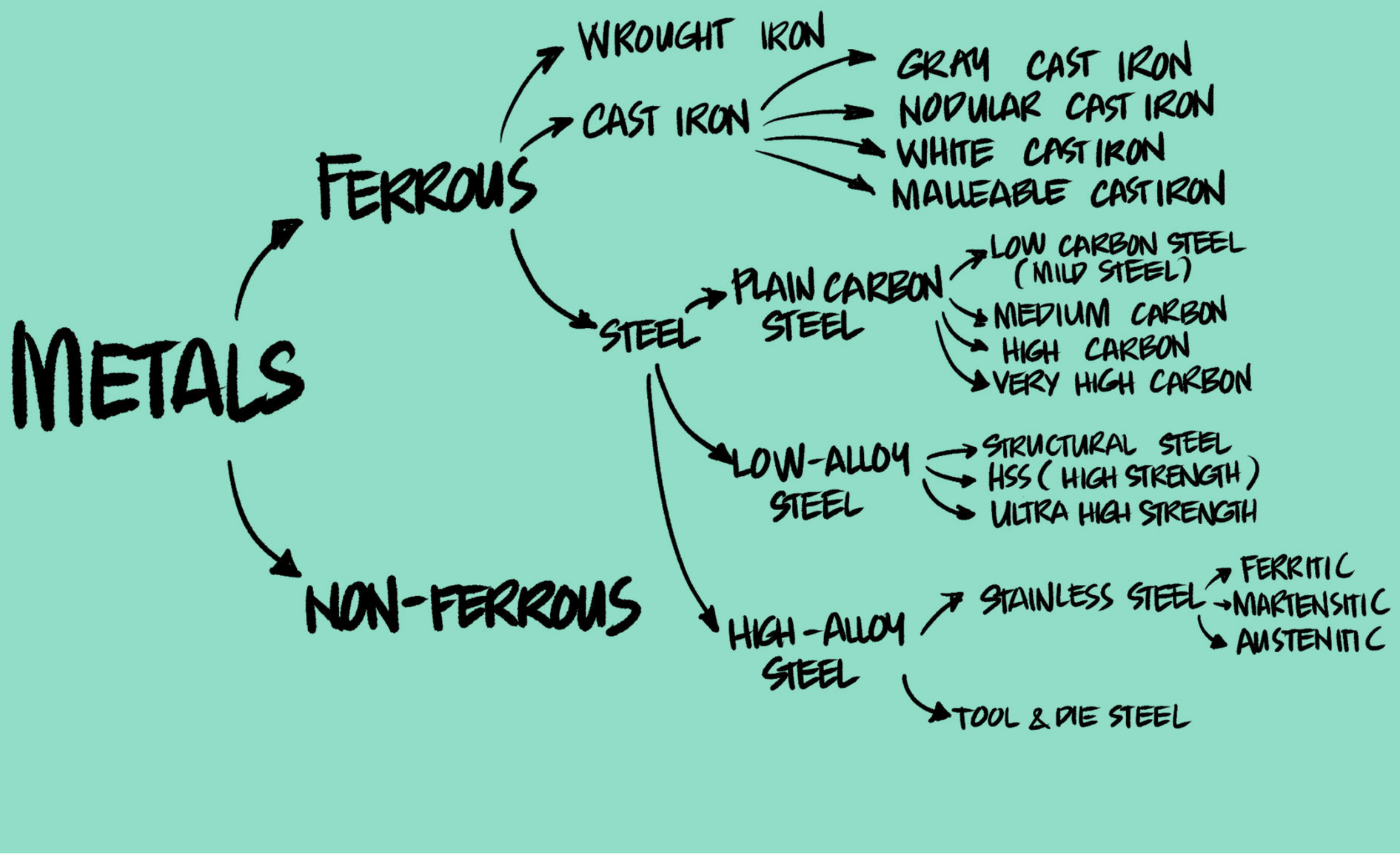
- IS THE MAXIMUM STRESS THAT A PART WILL BE PERMITTED TO SEE UNDER OPERATING CONDITIONS.

WORKING STRESS (ALSO KNOWN AS SAFE STRESS)

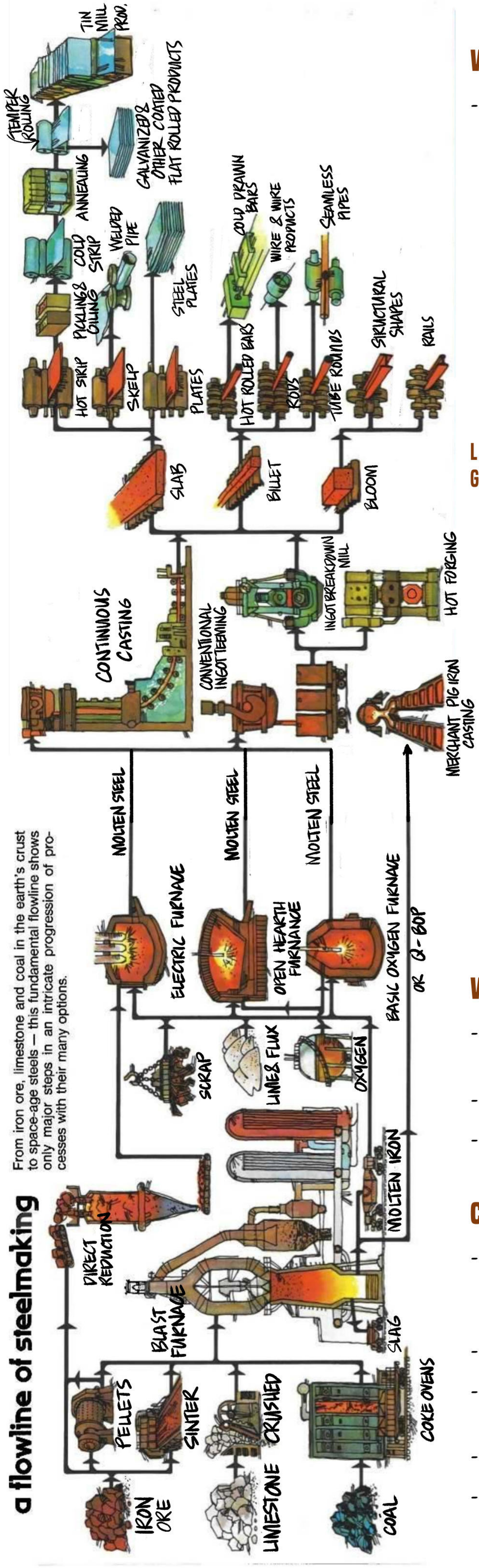
- IS USED INSTEAD OF DESIGN STRESS. WORKING STRESS IS BASED ON THE YIELD STRENGTH OF THE MATERIAL AND IS DEFINED AS

$$\sigma_W = \sigma_y / N$$

METALS



STEEL MAKING PROCESS



WHAT IS PIG IRON??

- IS THE **FIRST-STEP PURIFIED IRON** THAT IS CAST IN MOLDS. IT RESULTED AFTER THE IMPURITIES IN IRON ARE BURNED OUT IN A BLAST FURNACE. IT STILL CONTAIN A NUMBER OF ELEMENTS IN HIGH PROPORTIONS WHICH MAKE **PIG IRON BRITTLE.**

1 TON OF PIG IRON = 2 TONS OF IRON ORE + 1 TON OF COKE + ½ TON OF LIMESTONE

LIMESTONE IS USED AS A FLUX TO CARRY OFF THE GANGUE

ROUGH COMPOSITION OF PIG IRON	
CARBON	3.0 - 4.5%
MANGANESE	0.15 - 2.5%
PHOSPHORUS	0.1 - 2.0%
SILICON	1.0 - 3.0%
SULFUR	0.05 - 0.1%

FERROUS METALS

WROUGHT IRON

- WROUGHT IRON IS AN IRON ALLOY WITH A VERY LOW CARBON CONTENT IN CONTRAST TO CAST IRON.
- PRODUCED BY REPEATEDLY HEATING STRIPS
- EXPENSIVE, DISPERSES CARBON, SIMILAR TO PURE IRON

CAST IRON

- IS FORMED BY **REMELTING PIG IRON** AND OTHER MATERIALS, INCLUDING LIMESTONE AND PREPARED FLUXES IN A CUPOLA FURNACE.
- THEY DO NOT RUST EASILY
- IT IS BRITTLE AND GRAY IN COLOR. GOOD FOR CASTING COMPLEX SHAPES
- IT MELTS AT ABOUT **2200** F.
- IT CONTAINS ABOVE **2.14**WT%. C, AND IN PRACTICE HOWEVER, MOST CAST IRONS CONTAIN BETWEEN **3.0** AND **4.5**WT%C.

CLASSIFICATIONS OF CAST IRON

TYPE	DESCRIPTION	COMPOSITION	PROPERTY	ADD. INFO
WHITE	THE WHITE COLOR IS DUE TO FACT THAT IT HAS **NO GRAPHITE** AND WHOLE OF THE CARBON IS IN THE FORM OF **CARBIDE**. (KNOWN AS *CEMENTITE*) WHICH IS **THE HARDEST CONSTITUENT OF IRON.**	PEARLITE AND CEMENTITE	**HARD, BRITTLE**, NOT WELDABLE	IT BECOMES MALLEABLE CAST IRON **WHEN ANNEALED**
GREY	CAST IRONS **AT SLOW COOLING** CAST IRON WHICH IS SOFT GREY COLOR IS **BECAUSE OF THE PRESENCE OF CARBON IN THE FORM OF FREE GRAPHITE**	FERRITE AND DISPERSED GRAPHITE FLAKES	**EASILY MACHINED** AND WELDED	LEAST EXPENSIVE OF ALL METALLIC MATERIALS
MALLEABLE	PRODUCED BY HEAT TREATMENT OF WHITE C.I.	IRREGULAR GRAPHITE SPHEROIDS	**CAN BE WELDED, MACHINED** AND IS **DUCTILE** GOOD STRENGTH AND SHOCK RESISTANCE	TWO METHODS ARE USED: - BLACKHEART PROCESS - WHITEHEART PROCESS
NODULAR (DUCTILE)	GREY CAST IRONS + GRAPHITE PARTICLES MODIFIED BY MAGNESIUM BEFORE CASTING	SPHEROID NODULAR GRAPHITE PARTICLES IN FERRITE OR PEARLITE MATRIX	**DUCTILE AND MALLEABLE**	ALSO CALLED **HIGH STRENGTH CAST IRON**

STEEL

- STEEL IS AN ALLOY OF IRON AND CARBON. **SAMUEL HIGLEY OF SIMEBURY, CONNECTICUT**, PRODUCED THE FIRST STEEL IN AMERICA AROUND **1725**.

STEEL VS. IRON???

THE PRIMARY DIFFERENCE BETWEEN IRON AND STEEL IS THAT THE FORMER IS A METAL, WHEREAS THE LATTER IS AN ALLOY. **IRON IS SIMPLY A METAL ELEMENT** THAT OCCURS NATURALLY ON EARTH. IN COMPARISON, **STEEL IS A MAN-MADE ALLOY** THAT'S MADE BY MIXING IRON AND CARBON TOGETHER.

CLASSIFICATION OF COMMERCIAL STEELS

- PLAIN CARBON STEEL
- LOW-ALLOY STEEL
- HIGH-ALLOW STEEL

PLAIN CARBON STEEL

- HAS **1%** CARBON + TRACES OF MANGANESE, SULFUR, SILICON, AND PHOSPHORUS.

TYPE OF CARBON STEEL	% CARBON	APPLICATION
LOW CARBON	**<0.3**	WIRE, STRUCTURAL SHAPES, SCREW MACHINE
MEDIUM CARBON	**0.3-0.45**	AXLES, GEARS, AND SIMILAR PARTS
HIGH CARBON	**0.45-0.75**	DRILLS, CUTTING TOOLS, KNIVES
VERY HIGH CARBON	**0.75-1.5**	HARD STEEL PRODUCTS, METAL CUTTING TOOLS, TRUCK SPRINGS

INCREASING % OF CARBON CONTENT

- INCREASES **HARDNESS**
- INCREASES **STRENGTH**
- INCREASES **DIFFICULTY IN MACHINING AND WELDING**
- DECREASES **DUCTILITY**

LOW ALLOY STEELS

- CONTAINES LESS THAN **8%** OF ALLOYING ELEMENTS
- CARBON CONTENT IS BELOW **0.25%**

TYPES
STRUCTURAL STEEL
HIGH-STRENGTH STEEL
ULTRA HIGH – STRENGTH STEEL

HIGH ALLOY STEELS

- CONTAINES MORE THAN **8%** OF ALLOYING ELEMENTS
- CONTAINS AT LEAST **2** CHEMICAL ELEMENTS
- PROPERTIES DEPEND ON THE PERCENTAGE OF ELEMENTS
- HIGHLY CORROSION RESISTANT
- HIGH RELIABILITY

TYPES	
STAINLESS STEEL	FERRITIC
	MARTENSITIC
	AUSTENITIC
TOOL AND DIE STEELS	

STAINLESS STEELS

- FORMED BY ADDING **CHROMIUM (MINIMUM OF 10%)**
- RESISTANT TO STAINS, CORROSION, RUST, THAN ORDINARY STEEL

TYPES	COMPOSITION	ADD. INFO
FERRITIC	NO NICKEL **>12-27%** CHROMIUM	FOR TURBINE BLADES
MARTENSITIC	NO NICKEL HIGHER CARBON CONTENT	SURGICAL INSTRUMENTS
AUSTENITIC	**4-22%** NICKEL	GENERAL CORROSIVE PURPOSES, WELDABLE NON - MAGNETIC

DESIGNATION SYSTEM	
2XXX	**Cr-Ni-Mn** AUSTENITIC SS
3XXX	**Cr-Ni** AUSTENITIC SS
4XXX	**Cr** MARTENSITIC OR FERRITIC SS
5XXX	LOW **Cr** MARTENSITIC STEEL

TOOL AND DIE STEELS

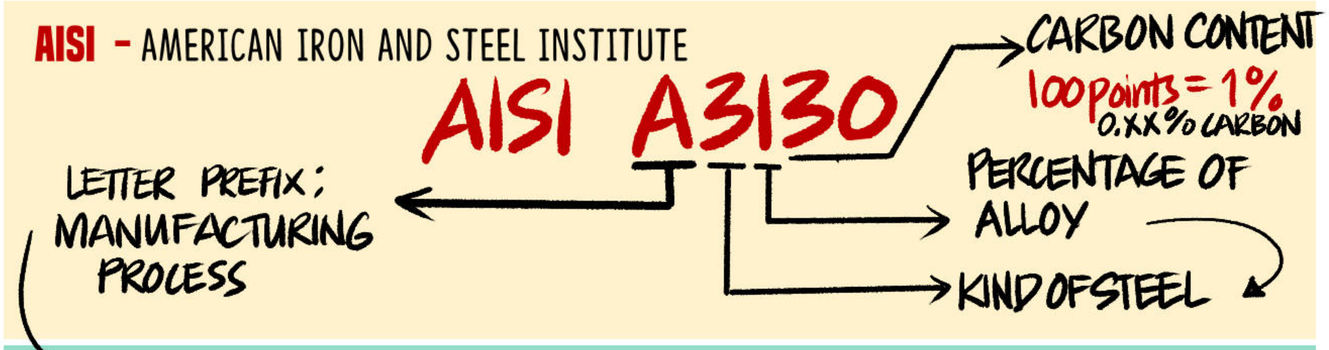
- HIGH CARBON AND ALLOY STEELS
- HIGH HARDNESS, STRENGTH, AND WEAR RESISTANCE
- **0.7% - 1.5%** CARBON
- USED TO **SHAPE** OTHER METALS

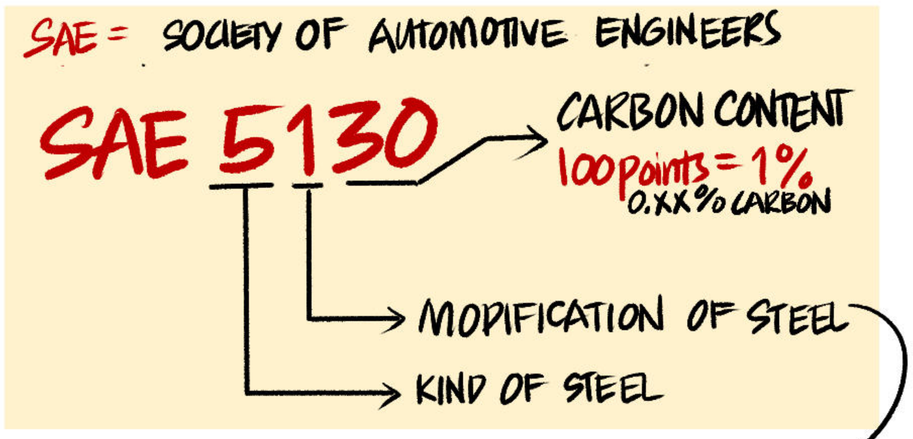
AISI AND SAE ONE LETTER SYSTEM: COMBINATION WITH A NUMBER OF TOOL STEEL	
W	**W**ATER HARDENED PLAIN CARBON STEEL
O	**O**IL HARDENING COLD WORK ALLOY STEEL
A	**A**IR
D	**D**IFFUSED
S	**S**HOCK RESISTANCE LOW CARBON TOOL STEEL
T	HIGH SPEED **T**UNGSTEN TOOL STEEL
M	HIGH SPEED **M**OLYBDENUM TOOL STEEL
H	**H**OT WORK TOOL STEEL
P	**P**LASTIC MOLD TOOL STEEL

TYPES OF FURNACES

OPEN HEARTH	CONVERTS PIG IRON INTO STEEL BY OXIDIZING **OUT THE SURPLUS CARBON** AND USING OXYGEN BY THROWING RAW IRON ORE INTO THE FURNACE
ELECTRIC FURNACE	SIMILAR TO OPEN HEARTH, DIFFERENCE IS THE METHOD OF **HEATING IS CHARGED**. USED IN PRODUCING QUALITY STEELS BECAUSE MELTING IS CONTROLLED CLOSELY.
CUPOLA	FURNACE IS CHARGED **WITH LAYERS OF COKE AND PIG IRON + SCRAP IRON**
BESSEMER	CONSISTS OF **BLOWING A STREAM OF AIR** THROUGH A MOLTEN MASS OF IRON PIG.

AISI AND SAE SPECIFICATION NUMBERS





LETTER PREFIX: MANUFACTURING PROCESS	
A	**A**LLOY, BASIC OPEN HEARTH
B	CARBON, ACID **B**ESSEMER
C	**C**ARBON, BASI**C** OPEN HEARTH
D	CARBON, ACI**D** OPEN HEARTH
E	**E**LECTRIC FURNACE STEEL

COMMON APPLICATION OF ALLOY STEELS (AISI)	
AISI 2330	BOLTS, STUDS, TUBING SUBJECTED TO TORSIONAL STRESSES.
AISI 2340	QUENCHED AND TEMPERED SHAFTING, CONNECTING RODS, VERY HIGHLY STRESSES BOLTS, FORGINGS
AISI 2350	HIGH-CAPACITY GEARS, SHAFTS, HEAVY DUTY MACHINE PARTS
AISI 3130	SHAFTS, BOLTS, STEERING KNUCKLES
AISI 3140	AIRCRAFT- AND TRUCK-ENGINE CRANKSHAFTS, OIL-WELL TOOL JOINTS, SPLINE SHAFTS, AXLES, EARTH MOVING EQUIPMENT
AISI 3150	WEAR-RESISTING PARTS IN EXCAVATING AND FARM MACHINERY, GEARS, FORGINGS
AISI 3240	SHAFTS, HIGHLY STRESSES PINS AND KEYS, GEARS
AISI 3300 SERIES	FOR HEAVY PARTS REQUIRING DEEP PENETRATION OF THE HEAT TREATMENT (HARDENABILITY) AND HIGH FATIGUE STRENGTH PER UNIT WEIGHT
AISI 4063	LEAF AND COIL SPRINGS
AISI 4130, 4140	AUTOMOBILE CONNECTING RODS AND AXLES, AIRCRAFT PARTS AND TUBING
AISI 4640	GEARS, SPLINED SHAFTS, HAND TOOLS, MISCELLANEOUS HEAVY DUTY MACHINE PARTS
AISI 8630	CONNECTING RODS, BOLTS, SHAPES; AIR HARDENS AFTER WELDING
AISI 8640, 8740	GEARS, PROPELLER SHAFTS, KNUCKLES, SHAPES

SECOND DIGIT: MODIFICATION OF STEEL	
0	PLAIN CARBON – (NOT MODIFIED)
1	RESULFURIZED
2	RESULFURIZED AND REPHOSPHERIZED
3	NON – RESULFURIZED, **Mn** OVER **10%**

NUMBER "1ST DIGIT"	STEEL
10XX	PLAIN CARBON
11XX	FREE CUTTING
13XX	MANGANESE
14XX	BORON
2XXX	NICKEL
3XXX	NICKEL CHROMIUM
303XX	HEAT AND COROSSION RESISTANT
4XXX	MOLYBDENUM
41XX	MOLYBDENUM-CHROMIUM
43XX	MOLYBDENUM-CHROMIUM-NICKEL
46XX	MOLYBDENUM-NICKEL
47XX	MOLYBDENUM-CHROMIUM-NICKEL
48XX	MOLYBDENUM-NICKEL
5XXX	CHROMIUM
514XX **515XX**	HEAT AND CORROSION RESISTANT
6XXX	CHROMIUM-VANADIUM
8XXX	NICKEL-CHROMIUM-MOLYBDENUM
92XX	SILICON-MANGANESE
9XXX	NICKEL-CHROMIUM-MOLYBDENUM (EXCEPT **92**XX)

EFFECTS OF ALLOYING ELEMENTS

ELEMENT	EFFECT
MANGANESE	- INCREASES ITS **HARDNESS, DUCTILITY, HIGH STRAIN HARDENING CAPACITY**, AND HAS EXCELLENT **WEAR RESISTANCE**, WHICH MAKE IT **IDEAL FOR IMPACT-RESISTING** TOOLS AND SIMILAR APPLICATIONS.
MANGANESE + SULFUR	PRODUCES SOFT MANGANESE SULFIDES, **WHICH PREVENTS THE FORMATION OF IRON SULFIDE.** • ***IRON SULFIDE*** WOULD COAT THE GRAIN BOUNDARIES AND IMPART BRITTLENESS.
SULFUR	• IS USUALLY **NOT DESIRED** IN STEEL BECAUSE OF THE EMBRITTLING EFFECT OF **IRON SULFIDE.** - IF THE PRODUCT IS MANGANESE SULFIDE, HOWEVER, THE SULFUR IS NOT HARMFUL (PROVIDED THAT THE SULFIDES ARE NOT IN LARGE QUANTITIES AND ARE WELL DISPERSED). - IF LARGE QUANTITIES OF MANGANESE SULFIDE ARE PRESENT IN THE PROPER FORM, THEY CAN IMPART DESIRABLE MACHINING PROPERTIES. **SOME FREE-MACHINING STEELS CONTAIN AS MUCH AS 0.08 TO 0.15% SULFUR IN COMBINATION WITH INCREASED MANGANESE CONTENT.**
NICKEL	- IS OFTEN ADDED TO INCREASE TOUGHNESS AND IMPACT RESISTANCE, PRIMARILY AT LOW TEMPERATURE. - AN IRON ALLOY WITH **36%** NICKEL, COMMONLY KNOWN AS **INVAR** , HAS A **NEAR-ZERO THERMAL EXPANSION COEFFICIENT** AND IS USED **FOR SENSITIVE MEASURING DEVICES.**
CHROMIUM	- AT LARGE PERCENTAGES CAN IMPART CORROSION RESISTANCE AND HEAT RESISTANCE, IN THE AMOUNTS USED IN LOW-ALLOY STEELS, THESE EFFECTS ARE RELATIVELY MINOR. - CHROMIUM, IN AMOUNTS THAT ARE USUALLY **LESS THAN 2%**, SERVES PRIMARILY TO INCREASE HARDENABILITY AND STRENGTH. - **1 CHROMIUM:2 NICKEL** . WHEN COMBINED WITH CARBON, THE CHROMIUM CARBIDES CAN ENHANCE WEAR RESISTANCE.
MOLYBDENUM	- USED IN ALLOY STEELS IN AMOUNTS **LESS THAN 0.3%** TO IMPROVE HARDENABILITY AND INCREASE STRENGTH PROPERTIES, PARTICULARLY UNDER DYNAMIC AND HIGH-TEMPERATURE CONDITIONS. - RESISTANCE TO TEMPER EMBRITTLEMENT IS ALSO ATTRIBUTED TO THE PRESENCE OF MOLYBDENUM. - EXTREMELY **STABLE AT ELEVATED TEMPERATURES** AND ARE USED TO RETAIN FINE GRAIN SIZE AND **PROVIDE STRENGTH AND CREEP RESISTANCE.** - USED IN HOT-WORK TOOL STEELS, SUCH AS THOSE USED IN FORGING DIES, TO IMPART HARDNESS THAT WILL PERSIST AT RED HEAT.
VANADIUM	- FORMS STRONG CARBIDES THAT PERSIST AT ELEVATED TEMPERATURE. - THUS **0.03 TO 0.25%** VANADIUM IN THE FORM OF CARBIDES CAN EFFECTIVELY INHIBIT GRAIN GROWTH AND INCREASE STRENGTH PROPERTIES (MOST NOTABLY ELASTIC LIMIT, YIELD POINT, AND IMPACT STRENGTH) WITH ALMOST NO LOSS OF DUCTILITY.
SILICON	- IT IS AN IMPORTANT ALLOYING ELEMENT IN CERTAIN HIGH-YIELD-STRENGTH STRUCTURAL STEELS AND IS ALSO USED IN **SPRING STEELS** (IN AMOUNTS OF ABOUT **2%**) AND TO PROMOTE THE LARGE GRAIN SIZE THAT IS DESIRABLE IN STEELS USED FOR MAGNETIC APPLICATIONS.
BORON	- IS A VERY POWERFUL HARDENABILITY AGENT ▪ **250 TO 750** TIMES AS EFFECTIVE AS **NICKEL** ▪ **75 TO 125** TIMES AS EFFECTIVE AS **MOLYBDENUM** ▪ ABOUT **100** TIMES AS POWERFUL AS **CHROMIUM**
PHOSPHORUS	- IS CONSIDERED UNDESIRABLE BECAUSE OF ITS EMBRITTLEMENT EFFECT. - IT INCREASES YIELD STRENGTH AND REDUCES DUCTILITY, AND IS BELIEVED TO INCREASE ATMOSPHERIC CORROSION WHEN USED WITH SMALL PERCENTAGES OF COPPER. - IF LIMITED, IT DOES IMPROVE MACHINABILIY.
TUNGSTEN	- FORMS STABLE CARBIDES AND IS ALSO USED AS A PRIMARY ALLOYING ELEMENT IN TOOL STEELS THAT MUST MAINTAIN THEIR HARDNESS AT HIGH ELEVATED TEMPERATURES. (FOR METAL CUTTING TOOLS)

CARBON	- AS THE **CARBON CONTENT IS INCREASED**, THE RESPONSE TO HEAT TREATMENT MORE PRONOUNCED AND THE STEEL'S HARDNESS AND TENSILE STRENGTH ARE INCREASED WHILE ITS DUCTILITY DECREASES.
LEAD	- WITH **0.15 TO 0.35%** ADDED IN STEELS WILL SHOW SUBSTANTIALLY IMPROVED MACHINABILITY, BUT THE ENVIRONMENTAL EFFECTS OF LEAD MAY LIMIT ITS ACCEPTABILITY
TITANIUM	- WHEN ADDED TO **18-8** STAINLESS STEEL WILL MAKE IT IMMUNE TO HARMFUL CARBIDE PRECIPITATION. - IT MAY BE ADDED TO LOW CARBON STEELS TO MAKE THEM SUITABLE TO PORCELAIN ENAMELING. - IT IS ALSO A CARBIDE FORMER.
ALUMINUM	- IS USED AS A DEOXIDIZER AND MAY BE ADDED TO STEELS IN AMOUNTS OF **0.95 TO1.30%** TO PRODUCE ALUMINUM NITRIDES DURING A NITRIDING OPERATION.
NIOBIUM / COLUMBIUM	- IS ALSO A CARBIDE FORMER. - IT HAS AN EFFECT SIMILAR TO TITANIUM IN THE **18-8** STAINLESS STEELS IN MAKING THE STEEL IMMUNE TO HARMFUL CARBIDE PRECIPITATION AND SUBSEQUENT INTERGRANULAR CORROSION.
BISMUTH, SENIUM & TELLURIUM	- ALLOY ELEMENTS THAT PROMOTES MACHINABILITY.
ZIRCUNIUM, CERIUM & CALCIUM	- CAN CONTROL THE SHAPE OF INCLUSIONS AND THEREBY PROMOTE TOUGHNESS.

NON-FERROUS METALS

COPPER AND COPPER ALLOYS

- THE BACKBONE OF ELECTRICAL INDUSTRY
- HEAVIER THAN IRON
  - HIGH ELECTRICAL & THERMAL CONDUCTIVITY
  - HIGH DUCTILITY
  - HIGH CORROSION RESISTANCE

1. ELECTROLYTIC TOUGH-PITCH COPPER

- REFINED COPPER CONTAINING BET. **0.02** AND **0.05%** OXYGEN. USED AS WIRE AND CABLE.

1. OXYGEN-FREE HIGH-CONDUCTIVITY COPPER

- MORE REFINED COPPER WITH MORE REDUCED OXYGEN CONTENT

CDA SYSTEM

COPPER DEVELOPMENT ASSOCIATION SYSTEM

- ADOPTED BY ASTM, SAE, AND US. GOV'T.

100-190	MOSTLY COPPER WITH **LESS THAN 2%** ALLOY
200-799	WROUGHT ALLOYS
800-900	CASTING ALLOYS

COPPER BASED ALLOYS

• COPPER-ZINC ALLOYS

BRASSES

- KNOWN RESULTING ALLOY OF COPPER & ZINC.
- WITH GOOD CORROSION RESISTANCE

CARTRIDGE BRASS	**70%** Cu + **30%** Zn
NAVAL BRASS	MUNTZ METAL + TIN

- DUCTILE & FORMABLE W/C INCREASES AS ZINC INCREASES.

• COPPER-TIN ALLOYS

BRONZES

- KNOWN RESULTING ALLOY OF COPPER AND **<12%** TIN
- USED TO DESIGNATE ANY COPPER ALLOY WHERE THE MAJOR ALLOY ADDITION IS NEITHER ZINC NOR NICKEL
- GOOD STRENGTH, TOUGHNESS, WEAR & CORROSION RESISTANCE

PHOSPHOR BRONZE	MOST POPULAR WROUGHT BRONZE

• COPPER-NICKEL ALLOYS

- HIGH THERMAL CONDUCTIVITY & STRENGTH
- HIGH STRESS CORROSION RESISTANCE, INCLUDING IN SEAWATER
- HEAT EXCHANGERS, COOKWARE, DESALINATION APPARATUS, COINAGE

TYPES	COMPOSITION
CUPRONICKEL	**2** TO **30%** NICKEL
NICKEL SILVERS	CONTAINS NO SILVER BUT **10** TO **30%** NICKEL AT LEAST **5%** ZINC
CONSTANTAN	**45%** NICKEL
MONEL	**67%** NICKEL

COPPER BASED ALLOYS (CONT.)

• ALUMINUM BRONZE ALLOYS

- HIGH STRENGTH, EXCELLENT CORROSION RESISTANCE
- **MARINE HARDWARE**, POWER SHAFTS, SLEEVE BEARINGS, PUMP & VALVE HANDLING **SEA WATER & VARIOUS INDUSTRIAL FLUIDS**

• SILICON BRONZES

- GOOD STRENGTH, FORMABILITY, **MACHINABILITY** AND CORROSION RESISTANCE TENSILE STRENGTH (**380MPA - 900MPA**)
- BOILER TANK, STOVE APPLICATIONS W/C REQUIRE A COMBINATION OF WELDABILITY, HIGH STRENGTH & CORROSION RESISTANCE

• COPPER-BERYLLIUM ALLOYS

- HAS SAME STRENGTH OF STEEL, **NON-SPARKING**, NON-MAGNETIC, AND NON-CONDUCTIVE
- CONTACT SPRINGS, **SPARK** RESISTANT **SAFETY** TOOLS AND SPOT WELDING ELECTRODES

ALUMINUM BASED ALLOYS

- RANKS **2ND** TO STEEL IN BOTH WORLDWIDE QUANTITY & EXPENDITURE
- THE MOST IMPORTANT OF THE NONFERROUS METALS.
- WEIGHS ONLY ONE-THIRD COMPARED TO STEEL
- E IS ONLY ONE-THIRD COMPARED TO STEEL
- WORKABILITY, LIGHT WEIGHT, CORROSION RESISTANCE, GOOD ELECTRICAL & THERMAL CONDUCTIVITY.

PURE ALU	VERY CORROSION RESISTANT DUE TO THE TIGHT, ADHERENT OXIDE COATING
ALU ALLOYS	OXIDE FORMATION IS RETARDED

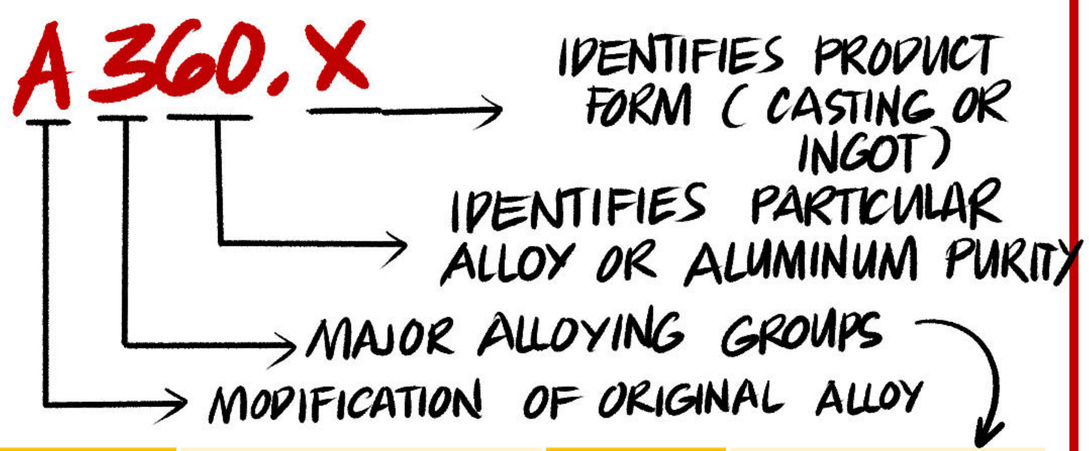
MAJOR CLASSIFICATIONS OF ALU ALLOYS:

1. WROUGHT ALLOYS
2. CASTING ALLOYS

CAST VS. WROUGHT??

CAST ALUMINIUM IS THE ALUMINIUM THAT **WAS MELTED** IN A FURNACE AND POURED INTO A MOLD. **WROUGHT** ALUMINIUM IS WHEN THE METAL IS **WORKED IN THE SOLID FORM** WITH THE HELP OF SPECIFIC TOOLS.

DESIGNATION SYSTEM: CAST ALU ALLOY


A360.X
IDENTIFIES PRODUCT FORM (CASTING OR INGOT)
IDENTIFIES PARTICULAR ALLOY OR ALUMINUM PURITY
MAJOR ALLOYING GROUPS
MODIFICATION OF ORIGINAL ALLOY

1XX.X	Al (99%+)	**5XX.X**	Mg
2XX.X	Cu	**7XX.X**	Zn
3XX.X	Si w/ Cu/Mg	**8XX.X**	Tin
4XX.X	Si	**9XX.X**	Other Element

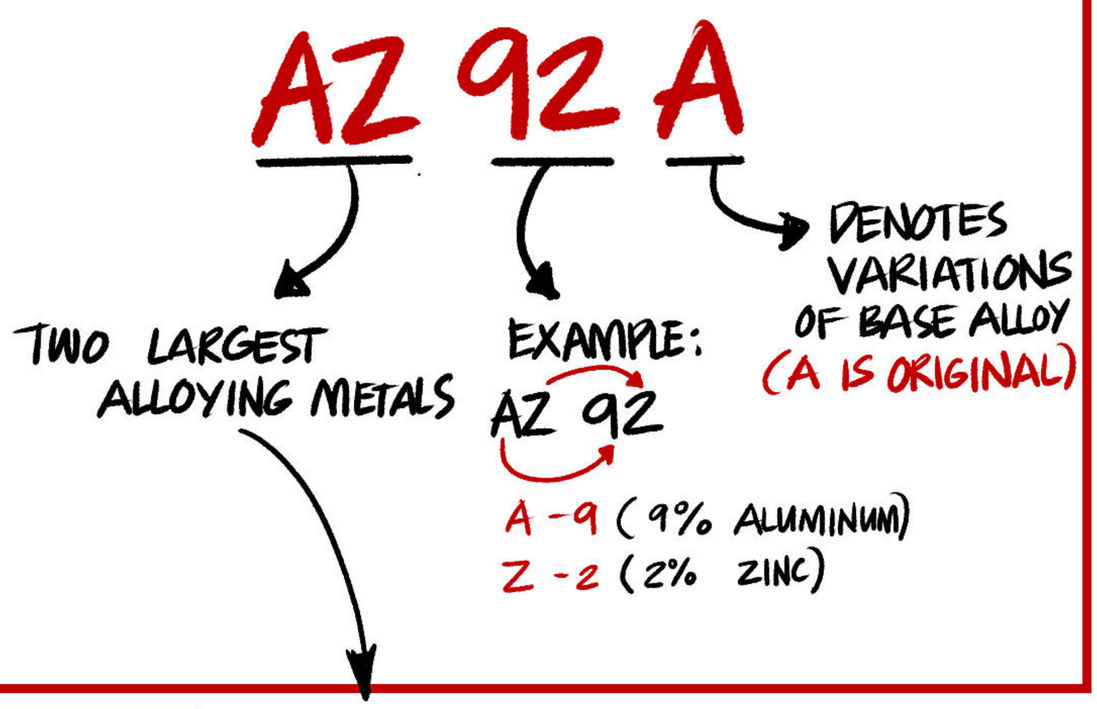
• ALUMINUM-LITHIUM ALLOYS

- HIGHER STRENGTH, GREATER STIFFNESS AND LIGHTER WEIGHT.
- EACH PERCENT OF LITHIUM OF LITHIUM (UP TO **4%**) REDUCES THE OVERALL WEIGHT BY **3%** AND INCREASES STIFFNESS BY **6%**.
- AN ATTRACTIVE AEROSPACE MATERIAL. IN COMMERCIAL AIRCRAFTS, IT CONSTITUTE **80%** OF THE WEIGHT

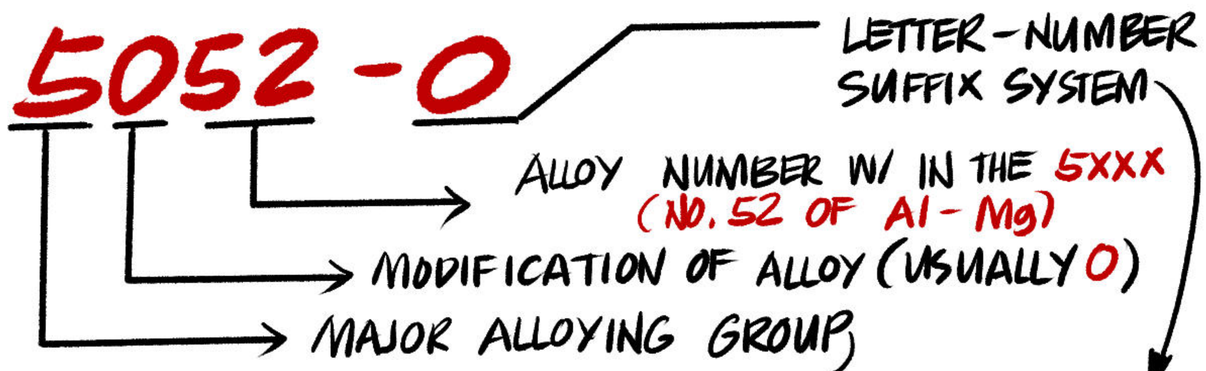
MAGNESIUM BASED ALLOYS

- IS THE LIGHTEST OF THE COMMERCIALLY IMPORTANT METALS.
- RELATIVELY **WEAK IN PURE STATE** AND IS ALMOST ALWAYS USED AS AN ALLOYING ELEMENT.

DESIGNATION SYSTEM: MAGNESIUM ALLOY



DESIGNATION SYSTEM: WROUGHT ALU ALLOY


5052 -O
LETTER-NUMBER SUFFIX SYSTEM
ALLOY NUMBER W/ IN THE 5XXX (NO. 52 OF Al - Mg)
MODIFICATION OF ALLOY (USUALLY O)
MAJOR ALLOYING GROUP

1XXX	Al (99%+)	**5XXX**	Mg
2XXX	Cu	**6XXX**	Mg & Si
3XXX	Mn	**7XXX**	Zn
4XXX	Si	**8XXX**	Other Element

F	FABRICATED
H	STRAINED HARDENED
O	ANNEALED
T	THERMALLY TREATED
W	SOLUTION HEAT TREATED

ALLOYING METALS

A	ALU	**H**	THORIUM	**Q**	SILVER
B	BISMUTH	**K**	ZIRCONIUM	**R**	CHROMIUM
C	COPPER	**L**	BERYLLIUM	**S**	SILICON
D	CADMIUM	**M**	MANGANESE	**T**	TIN
E	RARE EARTH	**N**	NICKEL	**Z**	ZINC
F	IRON	**P**	LEAD		

ZINC BASED ALLOY

- AS A PURE METAL, THE ONLY MAJOR USE OF ZINC IS THE GALVANIZING OF IRON & STEEL-AS SACRIFICIAL PLATING.
- MAY BE USED FOR EITHER **DIE CASTINGS OR SAND CASTINGS** FOR SUCH ARTICLES AS AUTOMOTIVE PARTS, BUILDING HARDWARE, PADLOCKS, TOYS, AND NOVELTIES.
- IT IS ALMOST AS HEAVY AS STEEL BUT WEAK AND BRITTLE.

TWO POPULAR ZINC DIE-CASTING ALLOYS:

1. AG40A ALSO KNOWN AS **ALLOY 903** OR ZAMAK 3
2. AC41A

TITANIUM BASED ALLOY

- IS STRONG, LIGHTWEIGHT, CORROSION-RESISTANT METAL.
- ITS PROPERTIES ARE BETWEEN THOSE OF STEEL & ALUMINUM.
- ITS DENSITY IS **56%** THAT OF STEEL AND ITS E RATIO ALSO **ABOUT ONE-HALF.**
    - AEROSPACE
    - CHEMICAL & ELECTROCHEMICAL PROCESSING EQUIPS.
    - MARINE IMPLEMENTS
    - BIOMATERIAL
    - REPLACEMENT OF STEEL WHERE WEIGHT SAVINGS IS DESIRED
    - REPLACEMENT OF ALUMINUM WHERE HIGH TEMP PERFORMANCE IS NECESSARY

NICKEL BASED ALLOY

- NOTED FOR ITS OUTSTANDING STRENGTH AND CORROSION RESISTANCE AT HIGH TEMPERATURES

MONEL	**67%** NICKEL + **30%** COPPER
- USED IN **CHEMICAL & FOOD PROCESSING** INDUSTRIES BECAUSE OF ITS OUTSTANDING CORROSION RESISTANCE TO SALT WATER, SULFURIC ACID, HIGH-VELOCITY & TEMP. STEAM. - USED FOR **TURBINE BLADES.**	
NICHROME	NICKEL AND CHROMIUM

LEAD, TIN & THEIR ALLOYS

LEAD

- IS USED IN STORAGE BATTERIES, CABLE CLADDING, AND SOUND-DAMPENING SHIELDS.

TIN

- USED AS A COATING ON STEEL TO IMPART CORROSION RESISTANCE.

	LEAD	TIN	ANTIMONY	COPPER	APPLICATION
TIN BABBIT	**85**	**5**	**8**	**2**	BEARINGS MATERIAL
LEAD BABBIT	**85**	**5**	**10**	**0.5**	SLOW SPEEDS AND MODERATE LOADS.
SOFT SOLDERS		**61.9**			

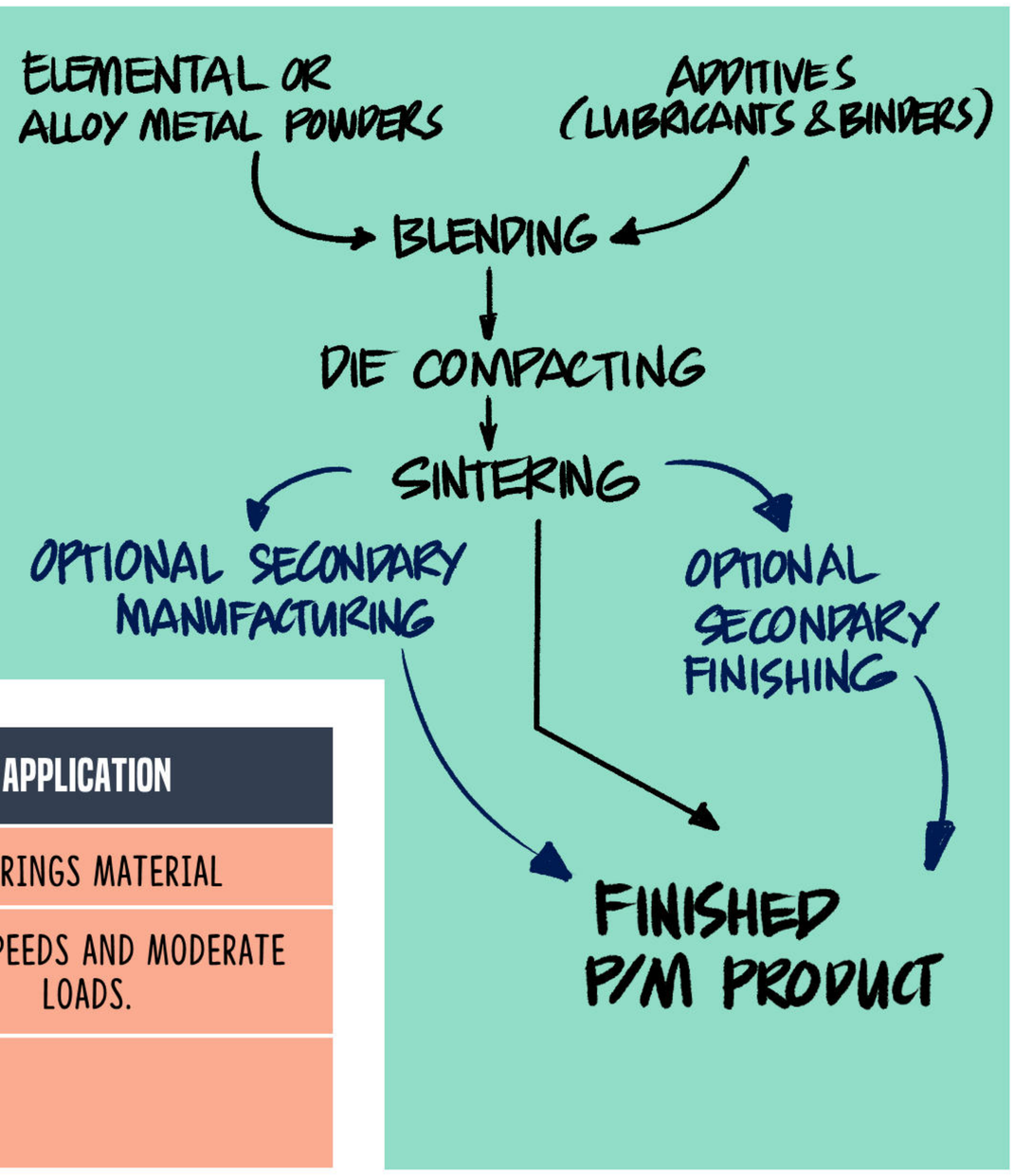
LESS-KNOWN METALS & ALLOYS

HAFNIUM **THORIUM** **BERYLLIUM**	- USED IN **NUCLEAR REACTORS** - LOW NEUTRON- ABSORPTION CHARACTERISTICS.
DEPLETED URANIUM	- USED IN COUNTER WEIGHTS & FLYWHEELS - VERY HIGH DENSITY
COBALT	- USED AS BASE METAL FOR SUPERALLOYS - IS USED AS A **BINDER** IN VARIOUS POWDER-BASED COMPONENTS AND SINTERED CARBIDES - **FOR RED HARDNESS**
ZIRCONIUM	- IS USED FOR ITS OUTSTANDING CORROSION RESISTANCE TO MOST ACIDS, CHLORIDES, & ORGANIC ACIDS. - ATTRACTIVE NEUTRON ABSORPTION CHARACTERISTICS. - SOME APPLICATIONS OF ZIRCONIUM ALLOYS INCLUDE HEAT EXCHANGERS, PROVIDE CLADDINGS, HEAT SHIELDS, AND ENGINE COMPONENTS.
RARE EARTH METALS	- HAVE BEEN INCORPORATED **INTO MAGNETS THAT OFFER INCREASED STRENGTH.**

GRAPHITE

TECHNICALLY NOT A METAL. IT INCREASES STRENGTH WHEN THE TEMPERATURE IS ELEVATED. IN LARGER QUANTITIES, IS USED AS ELECTRODES IN ARC FURNACES.

POWDER METALLURGY

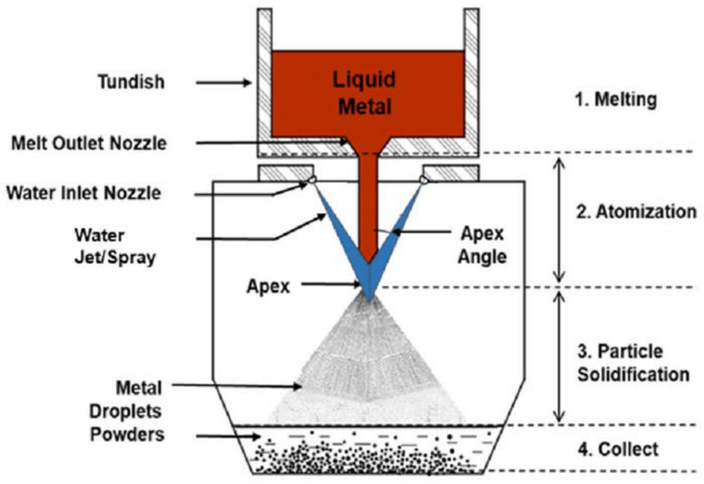


POWDER MANUFACTURE

METHODS USED FOR PRODUCING METAL POWDERS:

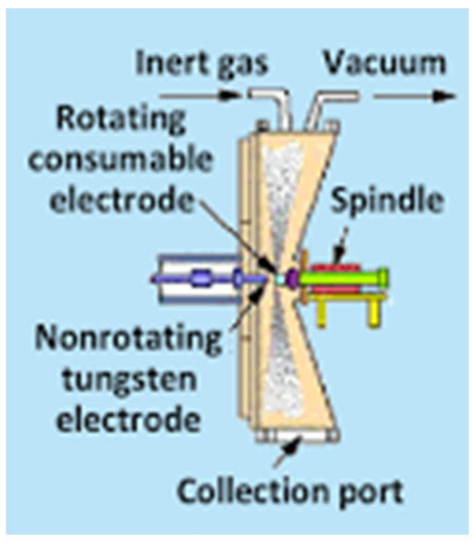
- **MELT ATOMIZATION**

- OVER **80%** OF ALL COMMERCIAL POWDERS IS PRODUCED BY MELT ATOMIZATION METHOD
- LIQUID IS FRAGMENTED INTO MOLTEN DROPLETS WHICH THEN SOLIDIFY INTO PARTICLES.
- IN ONE OF THE METHODS, THE MOLTEN METAL IS ATOMIZED BY A STREAM OF IMPACTING GAS OR LIQUID AS IT EMERGES FROM AN ORIFICE.



- **ATOMIZATION FROM A ROTATING CONSUMABLE ELECTRODE**

- ELECTRIC ARC IMPINGES ON A RAPIDLY ROTATING ELECTRODE WITH CENTRIFUGAL FORCE CAUSING THE MOLTEN DROPLETS TO FLY FROM THE SURFACE OF THE ELECTRODE.



POWDER MIXING AND BLENDING

- IT IS THE PROCESS OF MIXING TOGETHER THE METAL POWDERS AND LUBRICANTS (OR BINDERS) WITH ALLOY ADDITION UNTIL THEY ARE THOROUGHLY BLENDED.

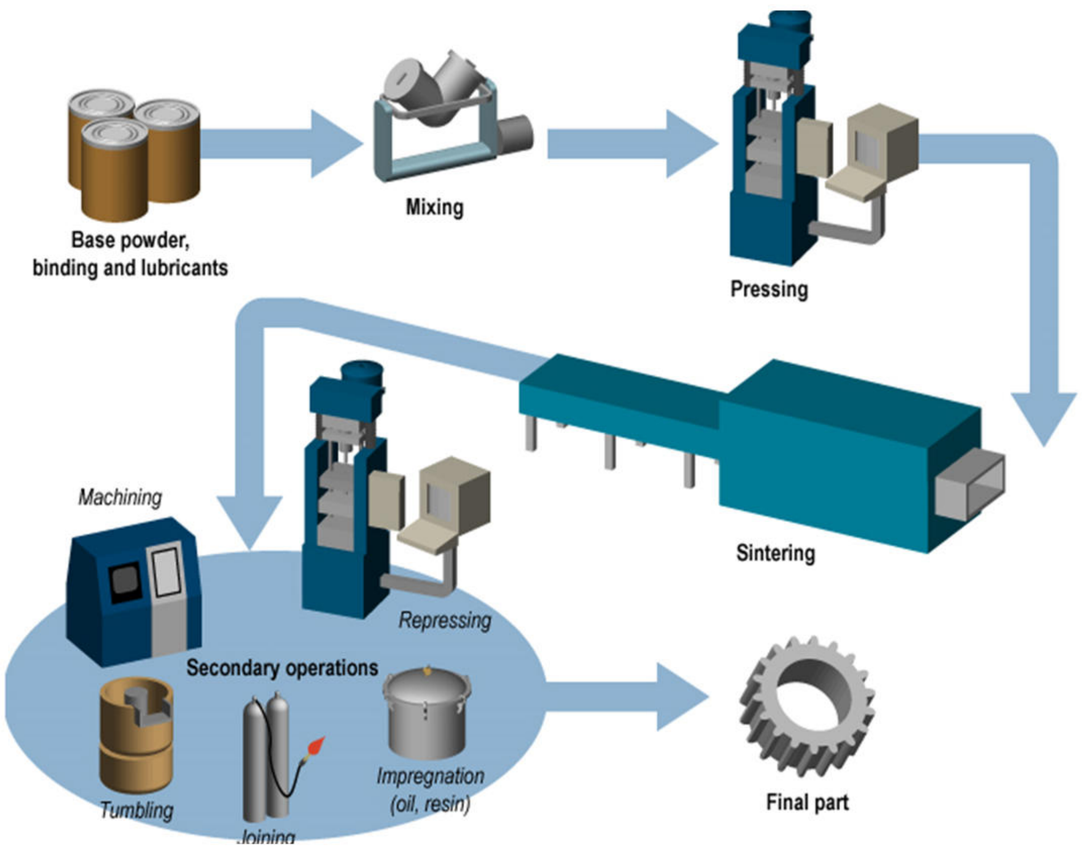
COMPACTING

- IN COMPACTING, THE BLENDED METAL POWDER IS FED INTO PRECISION DIE, COMPRESSED AND DENSIFIED INTO A SHAPE KNOWN AS GREEN COMPACT, USUALLY AT ROOM TEMPERATURE WITH PRESSURE RANGING FROM **3 TO 120 TONS/IN2** DEPENDING ON MATERIAL AND APPLICATION.

SINTERING

- IN THE SINTERING OPERATION, THE PRESSED-POWDER COMPACTS ARE HEATED IN A CONTROLLED-ATMOSPHERE ENVIRONMENT TO A TEMPERATURE BELOW THE MELTING POINT BUT HIGH ENOUGH TO PERMIT SOLID-STATE DIFFUSION, AND HELD FOR SUFFICIENT TIME TO PERMIT BONDING OF PARTICLES.

SINTERING OPERATION	
BURN OFF / PURGE STAGE	- COMBUST ANY AIR, VOLATILIZE AND REMOVE LUBRICANTS OR BINDERS THAT WOULD INTERFERE WITH GOOD BONDING, - SLOWLY RAISE THE TEMPERATURE IN A CONTROLLED MANNER
HIGH TEMPERATURE STAGE	- THE DESIRED SOLID-STATE DIFFUSION AND BONDING BETWEEN THE POWDER PARTICLES TAKE PLACE. - TIME MUST BE SUFFICIENT TO PRODUCE THE DESIRED DENSITY AND FINAL PROPERTIES, AND USUALLY VARIES **FROM 10 MINUTES TO SEVERAL HOURS.**
COOLING PERIOD	- REQUIRED TO **LOWER THE TEMPERATURE WHILE MAINTAINING THEM IN A CONTROLLED ATMOSPHERE.** - SERVES TO **PREVENT BOTH OXIDATION** THAT WOULD OCCUR UPON DISCHARGE INTO AIR AND **POSSIBLE THERMAL SHOCK** FROM RAPID COOLING.



NON-METALLIC MATERIALS

PLASTICS C_nH_{2n+2} → PARAFFIN-TYPE HYDROCARBON

- THE PLASTICS ARE SYNTHETIC MATERIALS WHICH ARE MOLDED INTO SHAPE UNDER PRESSURE WITH OR WITHOUT THE APPLICATION OF HEAT. $SG = 1.1 \text{ TO } 1.6$

1.THERMOPLASTICS

-CAN BE MELTED REPEATEDLY

2. THERMOSETTING PLASTICS

-ONCE SHAPED, CANNOT BE MELTED

TYPES OF THERMOPLASTICS

ABS	- LOW WEIGHT, GOOD STRENGTH AND **VERY TOUGH** - DIMENSIONALLY **STABLE BUT FLAMMABLE**
ACRYLICS (LUCITE & PLEXIGLASS)	- **HIGHEST OPTICAL CLARITY** TRANSMITTING **90%** OF LIGHT - DIELECTRIC STRENGTHS; RESIST WEATHERING; STRETCH RATHER EASILY.
CELLULOSE ACETATE	- GOOD INSULATING QUALITIES - EASILY MOLDED - HIGH **MOISTURE ABSORPTION** IN MOST GRADES AND AFFECTED BY ALCOHOLS AND ALKALIES
CELLULOSE ACETATE BUTYRATE	- **HIGH IMPACT STRENGTH & MOISTURE RESISTANCE THAN CELLULOSE ACETATE** - WITHSTAND ROUGHER USAGE.
ETHYL CELLULOSE	- HIGH ELECTRICAL RESISTANCE & IMPACT STRENGTH - RETAINS TOUGHNESS AT LOW TEMPERATURES
FLUOROCARBONS	- **VERY LOW COEFFICIENT OF FRICTION (TEFLON)** - USED FOR NON-LUBRICATING BEARINGS AND **NONSTICK COATINGS FOR COOKING UTENSILS AND ELECTRICAL IRONS.**
POLYAMIDES (NYLON)	- LOW COEFFICIENT OF FRICTION; GOOD STRENGTH, **ABRASION RESISTANCE**, AND TOUGHNESS - EXCELLENT DIMENSIONAL STABILITY - USED FOR BEARINGS AND AS MONOFILAMENTS FOR **TEXTILES, FISHING LINE, AND ROPES.**
POLYCARBONATES	- HIGH STRENGTH & OUTSTANDING TOUGHNESS, GOOD DIMENSIONAL STABILITY
POLYETHYLENES	- INEXPENSIVE, TOUGH, GOOD CHEMICAL RESISTANCE & HIGH ELECTRICAL RESISTANCE; - **SUBJECT TO WEATHERING VIA ULTRAVIOLET LIGHT; FLAMMABLE** - **USED FOR GROCERY BAGS, MILK JUGS, TUBES, PIPES, SHEETING, AND ELECTRICAL WIRE INSULATION.**
POLYSTYRENES (STYROFOAM)	- HIGH DIMENSIONAL STABILITY & LOW WATER ABSORPTION - BEST ALL-AROUND DIELECTRIC - **CLEAR, HARD, & BRITTLE** - OFTEN USED FOR **RIGID PACKAGING**, CAN BE FOAMED TO PRODUCE EXPANDED POLYSTYRENE; BURNS READILY
VINYLS (SARAN AND TYGON)	- TEAR RESISTANT - WATER RESISTANT IN **RIGID FORMS** - **USED FOR FLOOR AND WALL COVERING, UPHOLSTERY FABRICS** - LIGHTWEIGHT WATER HOSE

TYPES OF THERMOSETS

EPOXIES	- ELASTICITY, CHEMICAL RESISTANCE, MOISTURE RESISTANCE - USED AS **ADHESIVES, BONDING AGENTS, COATINGS**, AND IN FIBER LAMINATES
MELAMINES	- EXCELLENT RESISTANCE TO HEAT, WATER, AND MANY CHEMICALS - FULL RANGE OF TRANSLUCENT AND OPAQUE COLORS - **USED IN TREATING PAPER AND CLOTH TO IMPART WATER REPELLENT PROPERTIES.**
PHENOLICS	- HARD, STRONG, LOW COST, EASILY MOLDED BUT BRITTLE - RESISTANT TO HEAT AND MOISTURE; OPAQUE, BUT WITH A WIDE COLOR RANGE; WIDE VARIETY OF FORMS; SHEET, ROD, TUBE, AND LAMINATE.
POLYESTERS	- STRONG & RESIST ENVIRONMENTAL INFLUENCES WELL - **USES INCLUDE BOAT & CAR BODIES, PIPE, VENTS & DUCTS, TEXTILES, ADHESIVES, COATINGS, AND LAMINATES.**
SILICONES	- **SEMIORGANIC SPINE MOLECULES** WITH ALTERNATING SILICON & OXYGEN ATOMS - HEAT & WEATHER RESISTANT - HIGH DIELECTRIC PROPERTIES - **EXCELLENT SEALANTS.**
UREA-FORMALDEHYDE	- **PROPERTIES SIMILAR TO PHENOLICS BUT AVAILABLE IN LIGHTER COLORS** - USEFUL IN **CONTAINERS & HOUSINGS BUT NOT OUTDOORS** - USED IN **LIGHTING FIXTURES** BECAUSE OF TRANSLUCENCE IN THIN SECTIONS.

ADDITIVES:

FILLRS

- **WOOD FLOUR** – FINE SAWDUST, GENERAL PURPOSE, GOOD MOLDABILITY
- **CLOTH FIBERS** – IMPROVE IMPACT STRENGTH, FAIR MOLDABILITY
- **MACERATED CLOTH** – IMPROVE IMPACT STRENGTH, LIMITED MOLDABILITY
- **GLASS FIBERS** – TRANSLUCENCE
- **MICA** – ELECTRICAL PROP., LOW MOISTURE ABSORPTION
- **CALCIUM CARBONATE, SILICA, TALC, AND CLAY** – SERVE AS EXTENDERS

COLORING AGENTS

- **DYES** – SOLUBLE IN THE RESINS, USED FOR TRANSPARENT
- **PIGMENTS** – INSOLUBLE IN THE RESINS, USED FOR OPAQUE

PLASTICIZERS

LUBRICANTS – WAXES, STEARATES, SOAPS

STABILIZERS

ANTIOXIDANTS

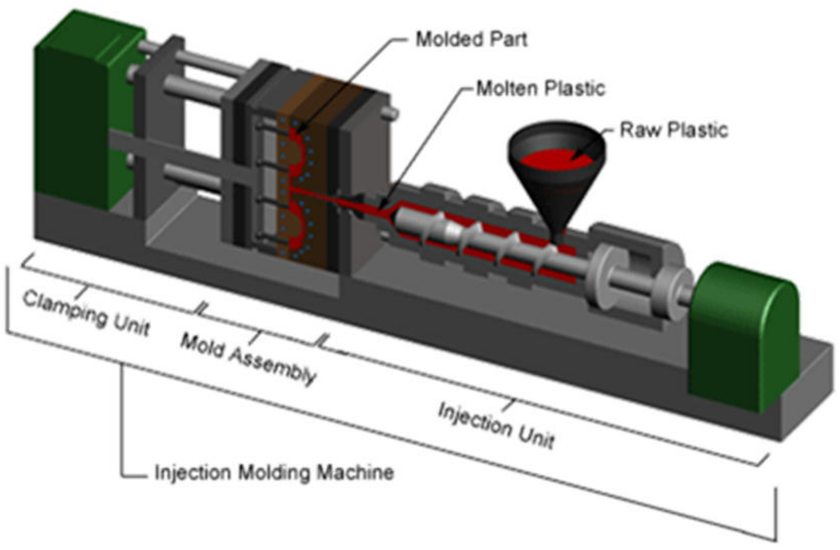
FLAME RETARDANTS

MANUFACTURING OF PLASTIC PRODUCTS

FORMING THERMOPLASTICS

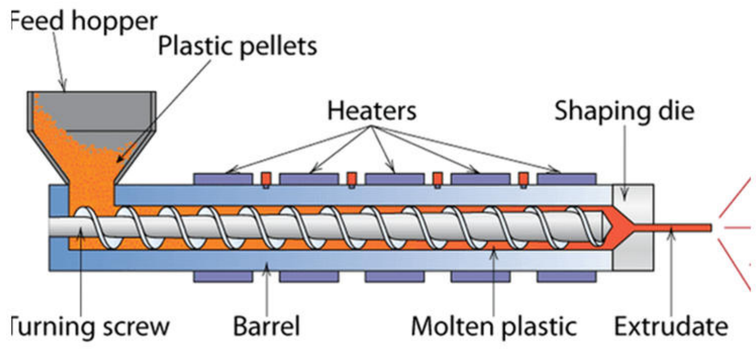
INJECTION MOLDING

- USES A PISTON OR SCREW TO FORCE PLASTIC RESIN THROUGH A **HEATED TUBE INTO A MOLD**, WHERE THE PLASTIC COOLS AND HARDENS TO THE SHAPE OF THE MOLD.
- THERMOPLASTIC ITEMS MADE BY INJECTION MOLDING INCLUDE TOYS, COMBS, CAR GRILLS, AND VARIOUS CONTAINERS.



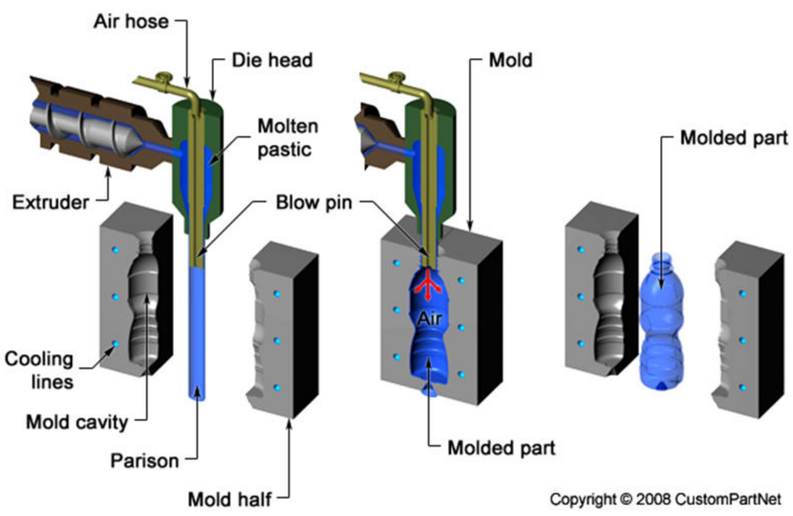
EXTRUSION

- PLASTIC PELLETS ARE FIRST HEATED IN A LONG BARREL.
- SIMILAR TO THAT OF **A PASTA-MAKING OR SAUSAGE-STUFFING** MACHINE, A ROTATING SCREW THEN FORCES THE HEATED PLASTIC THROUGH A DIE.
- PRODUCTS INCLUDE **GARDEN HOSES, DRINKING STRAWS, PIPES, AND ROPES**.
- FORCED THROUGH EXTREMELY FINE DIE HOLES CAN BE COOLED AND **WOVEN INTO FABRICS FOR CLOTHES, CURTAINS, AND CARPETS.**



BLOW MOLDING

- USED **TO FORM BOTTLES** AND OTHER CONTAINERS FROM SOFT, HOLLOW THERMOPLASTIC TUBES.
- MAKE MANY PLASTIC CONTAINERS, INCLUDING SOFT-DRINK BOTTLES, JARS, DETERGENT BOTTLES, AND STORAGE DRUMS.

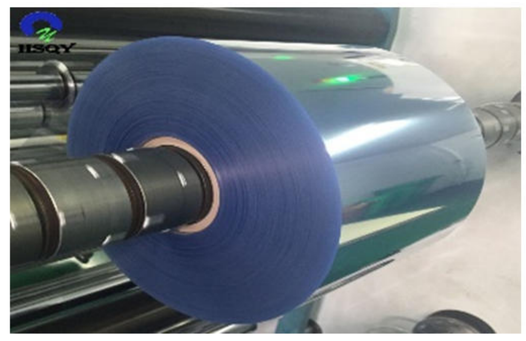


BLOW FILM EXTRUSION

- USES AN INDUSTRIAL BLOWER TO **EXPAND A HOT PLASTIC TUBE INTO A LIGHT, STRONG PLASTIC BAG.** THE AIR INFLATES THE PLASTIC TUBE LIKE A BALLOON, UNTIL A BAG WITH THE DESIRED PROPERTIES IS FORMED.
- BLOW FILM EXTRUSION IS THE PROCESS **USED TO MAKE PLASTIC GROCERY, GARBAGE BAGS AND CONTINUOUS SHEETS.**

CALENDERING

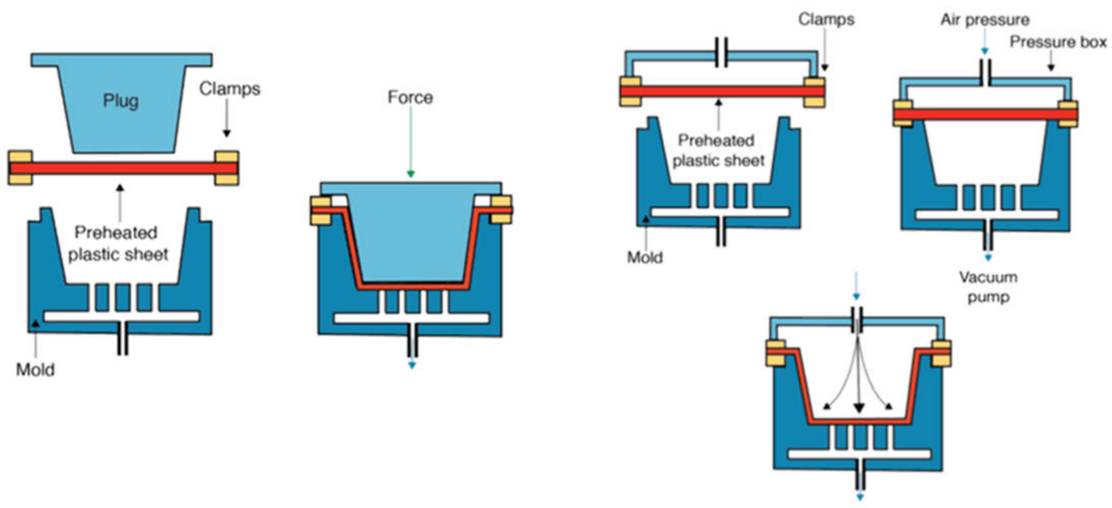
- PAPER, CLOTH, AND OTHER PLASTICS MAY BE **PRESSED BETWEEN LAYERS OF CALENDERED PLASTIC** TO MAKE ITEMS SUCH AS **CREDIT CARDS, PLAYING CARDS, AND WALLPAPER.**



THERMOFORMING

- PRODUCTS MADE FROM THERMOFORMED SHEETS **INCLUDE TRAYS, SIGNS, BRIEFCASE SHELLS, REFRIGERATOR DOOR LINERS, AND PACKAGES**

PRESSURE VS. VACUUM FORMING??

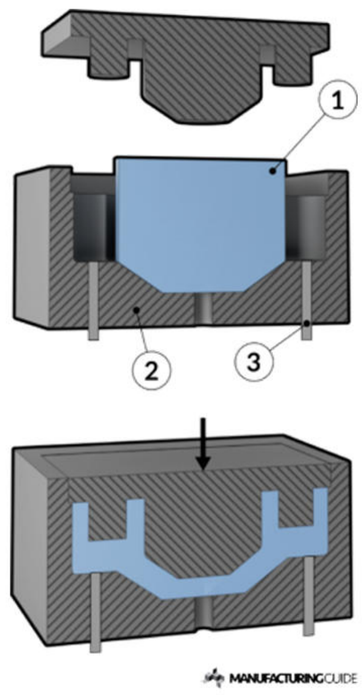


MANUFACTURING OF PLASTIC PRODUCTS

FORMING THERMOSETS

COMPRESSION MOLDING

- CREATES HARD, HEAT-RESISTANT PLASTIC PRODUCTS, INCLUDING DINNERWARE, TELEPHONES, TELEVISION SET FRAMES, AND ELECTRICAL PARTS.

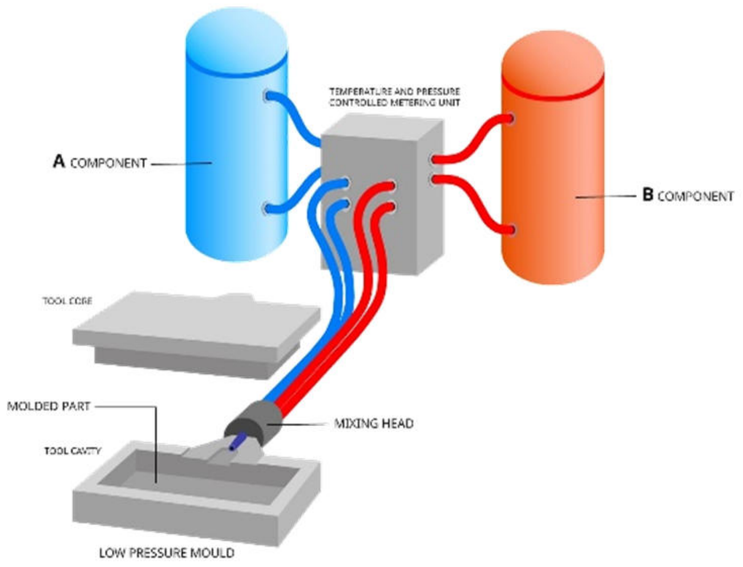


LAMINATING

- USED AS A **SURFACE FINISH FOR FURNITURE**, AND KITCHEN AND BATHROOM COUNTERTOPS.
- **ELECTRIC CIRCUIT BOARDS** ARE ALSO LAMINATED FROM RESIN-IMPREGNATED PAPER, FABRIC, AND GLASS FIBERS.

REACTION INJECTION MOLDING (RIM)

- STRONG, SIZABLE, AND DURABLE PLASTIC PRODUCTS SUCH AS **AUTOMOBILE BODY PANELS, SKIS, AND BUSINESS MACHINE HOUSINGS**
- LIQUID THERMOSETTING RESIN IS COMBINED WITH A **CURING AGENT** *(A CHEMICAL THAT CAUSES THE POLYMER MOLECULES TO CROSS-LINK)* AND INJECTED INTO A MOLD. MOST PRODUCTS MADE BY REACTION INJECTION MOLDING ARE **MADE FROM POLYURETHANE.**



ELASTOMERS

- ARE SPECIAL CLASS OF **LINEAR POLYMERS THAT DISPLAY AN EXCEPTIONALLY LARGE AMOUNT OF ELASTIC DEFORMATION WHEN FORCE IS APPLIED.**

NATURAL RUBBER

- IS THE OLDEST COMMERCIAL ELASTOMER FROM THE **PROCESSED SAP OF THE TROPICAL TREE.**
- IT SHOULD NOT BE USED AT TEMPS ABOVE **800 C**. OUTSTANDING IN FLEXIBILITY, GOOD ELECTRICAL INSULATOR, RESISTANT TO MOST INORGANIC ACIDS, SALTS, AND ALKALIES. **BUT HAVE POOR RESISTANCE TO PETROLUEM PRODUCTS.**
- DISCOVERED BY **CHARLES GOODYEAR.**

SYNTHETIC ELASTOMERS

- ARE ARTIFICIAL ELASTOMERS DEVELOPED TO OVERCOME SOME OF THE LIMITATIONS OF NATURAL RUBBER, WHICH NOW HAVE DISTINCTLY DIFFERENT AND, FREQUENTLY, SUPERIOR PROPERTIES.

1.POLYISOPRENE
2. SILICONE RUBBERS

CERAMICS

- CERAMICS ARE **COMPOUNDS OF METALLIC & NONMETALLIC** ELEMENTS **(OFTEN IN FORMS OF OXIDES, CARBIDES AND NITRIDES).**

MANUFACTURE OF CERAMICS

1. MOLDING

A. SLIP CASTING IS USED TO FORM HOLLOW CERAMIC OBJECTS
B. PRESSURE CASTING
C. INJECTION MOLDING USES PISTON TO MAKE SMALL, INTRICATE OBJECTS
D. EXTRUSION USES ROTATING SCREWS TO FORCE THE HEATED MATERIAL THROUGH AN OPENING.

2. DENSIFICATION

REFRACTORY MATERIALS

- ARE CERAMICS THAT HAVE BEEN DESIGNED TO **PROVIDE ACCEPTABLE MECHANICAL OR CHEMICAL PROPERTIES WHILE AT HIGH TEMPERATURES.**

ACIDIC REFRACTORIES

ARE BASED ON **SILICON DIOXIDE AND ALUMINUM OXIDE** AND CAN BE COMPOUNDED TO PROVIDE HIGH TEMP RESISTANCE ALONG W/ HIGH HARDNESS & GOOD MECH'L PROPERTIES. **(MACHINABLE SILICA CERAMICS HAVE BEEN USED AS INSULATING TILES ON THE US SPACE SHUTTLE.)**

BASIC REFRACTORIES

ARE BASED ON **MAGNESIUM OXIDE**.

NEUTRAL REFRACTORIES

CONTAINS **CHROMIUM (III) OXIDE** TO SEPARATE THE BASIC & ACIDIC MATERIALS SINCE THEY TEND TO ATTRACT ONE ANOTHER.

ABRASIVE

SILICON CARBIDE & ALUMINA

- OFTEN USED FOR ASBRASIVE APPLICATIONS, SUCH AS GRINDING.

INDUSTRIAL DIAMOND & CUBIC BORON NITRIDE

- ARE CALLED SUPERABRASIVES.

CERAMICS FOR ELECTRICAL & MAGNETIC APPLICATIONS

SILICON CARBIDE

- USED AS RESISTORS AND HEATING ELEMENTS FOR **ELECTRIC FURNACES**.

BARIUM TITANATE

- USED IN **CAPACITORS AND TRANSDUCERS**

ALUMINUM OXIDE

- MAKE EXCELLENT **HIGH VOLTAGE INSULATORS**

GLASSES

- ARE BASED ON **SILICA (SiO_2)** WITH ADDITIVES TO ALTER THE STRUCTURE OR REDUCE THE MELTING POINT.

GLASS CERAMICS

- ARE FIRST SHAPED AS A GLASS AND THEN **HEAT TREATED** TO PROMOTE RECRYSTALLIZATION OF THE MATERIAL. **ITS STRENGTH IS GREATER THAN THE TRADITIONAL GLASSES, AND THE THERMAL EXPANSION COEFFICIENT IS NEAR ZERO.**
- **PYROCERAM MATERIAL** —FOUND IN CORNINGWARE

STRUCTURAL CERAMICS

SILICON CARBIDE & SILICON NITRIDE

- OFFER EXCELLENT STRENGTH AND **WEAR RESISTANCE W/ MODERATE TOUGHNESS.**
- APPLICATIONS IS IN **TURBINE BLADES.**

PARTIALLY STABILIZED ZIRCONIA

- COMBINES THE **RESISTANCE TO THERMAL SHOCK, WEAR, CORROSION**
- LOW THERMAL CONDUCTIVITY AND LOW FRICTION COEFFICIENT WITH THE ENHANCED STRENGTH AND TOUGHNESS.

TRANSFORMATION-TOUGHENED ZIRCONIA

- HAS **EVEN GREATER TOUGHNESS**

SIALON (SI-AL-O-N STRUCTURAL CERAMIC)

- IS **STRONGER THAN STEEL**, EXTREMELY HARD **AND AS LIGHT AS ALUMINUM.**

ADVANCED CERAMICS AS CUTTING TOOLS

SILICON CARBIDE

- COMMON IN GRINDING TOOLS

COBALT-BONDED TUNGSTEN CARBIDE

- **POPULAR ALTERNATIVE TO HIGH SPEED TOOL STEELS** FOR MANY CUTTING APPLICATIONS.

SILICON NITRIDE

CUBIC BORON NITRIDE

POLYCRYSTALLINE DIAMOND CUTTING TOOLS

ALUMINUM OXIDE REINFORCED W/ SILICON CARBIDE

- OFFER GREATER TOOL LIFE, HIGHER CUTTING SPEEDS, AND REDUCED MACHINE DOWNTIME.

BIOCERAMICS

POROUS ALUMINA

- IS USED FOR **HIP JOINTS, DENTAL CAPS, DENTAL BRIDGES.**

CALCIUM HYDROXYL PHOSPHATE

- IS USED TO RECONSTRUCT **FRACTURED OR DISEASED BONE.**

NUCLEAR POWER

URANIUM CERAMIC PELLETS

- ARE USED TO GENERATE NUCLEAR POWER.

COMPOSITES

- ARE HETEROGENEOUS SOLIDS CONSISTING OF TWO OR MORE DIFFERENT MATERIALS THAT ARE **MECHANICALLY OR METALLURGICALLY BONDED** TOGETHER.

3 DISTINCT FAMILIES OF COMPOSITES

LAMINAR OR LAYERED COMPOSITES	- PLYWOOD, SAFETY GLASS
PARTICULATE COMPOSITES	- CONCRETE & ASPHALT
FIBER-REINFORCED COMPOSITES	- FIBER GLASS

ADVANCED FIBER-REINFORCED COMPOSITES

DEVELOPED FOR APPLICATIONS REQUIRING EXCEPTIONAL COMBINATIONS OF STRENGTH, STIFFNESS, AND LIGHT WEIGHT.

FOUR BASIC TYPES OF ADVANCED COMPOSITES:

- ADVANCED ORGANIC OR RESIN-MATRIX COMPOSITES
- METAL-MATRIX COMPOSITES
- CARBON-CARBON COMPOSITES
- CERAMIC-MATRIX COMPOSITES

CORROSION & DEGRADATION OF MATERIALS

ELECTROCHEMICAL PROCESS

- CHEMICAL REACTION INVOLVING THE TRANSFER OF ELECTRONS AT THE METAL SURFACE

$Zn \longrightarrow Zn^{2+} + 2e^-$ (OXIDATION REACTION, ANODE)

$2H^+ + 2e^- \longrightarrow H_2$ (REDUCTION REACTION, CATHODE)

ELECTROLYTE

SOLUTION CAPABLE OF **CONDUCTING ELECTRIC** CURRENT VIA THE MOVEMENT OF POSITIVE OR NEGATIVE IONS

- ANION: NEGATIVE ION
- CATION: POSITIVE ION

ANODE:

ATOMS **LOSE** ELECTRONS TO THE EXTERNAL CIRCUIT, **BECOMING + IONS**; WHERE **OXIDATION TAKES PLACE.**

CATHODE:

IONS **GAIN** ELECTRONS FROM THE EXTERNAL CIRCUIT; WHERE **REDUCTION TAKES PLACE**

CORROSION ON METALS

UNIFORM CORROSION

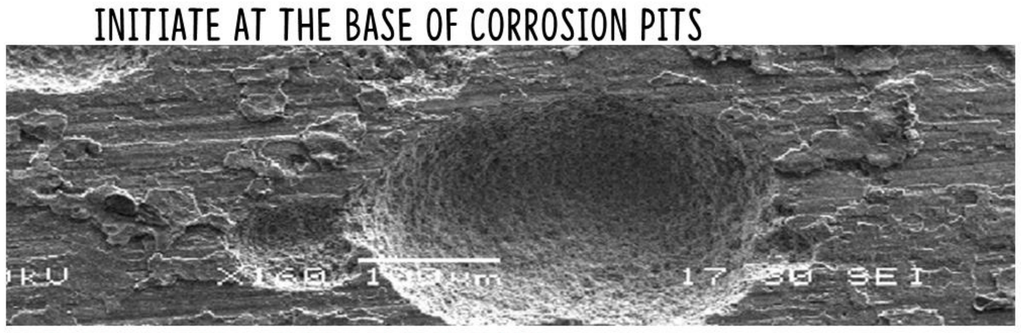
- CHARACTERIZED BY CORROSIVE ATTACK PROCEEDING **EVENLY OVER THE ENTIRE SURFACE AREA**, OR A LARGE FRACTION OF THE TOTAL AREA.

GALVANIC CORROSION

- TWO DIFFERENT METALS
- (ELECTRICALLY CONNECTED AND IN THE SAME ELECTROLYTE)
- **MORE ANODIC METAL CORRODES FASTER**
- **LESS ANODIC METAL CORRODES SLOWER (IS PROTECTED)**

PITTING CORROSION

- APART FROM THE LOCALIZED **LOSS OF THICKNESS**
- CORROSION PITS CAN ALSO BE HARMFUL BY ACTING AS **STRESS RISERS**. FATIGUE AND STRESS CORROSION CRACKING MAY INITIATE AT THE BASE OF CORROSION PITS



CREVICE CORROSION

- CREVICE CORROSION OCCURS AT SHIELDED AREAS THAT CONTAIN A SMALL VOLUME OF AQUEOUS SOLUTION.

INTERGRANULAR CORROSION

- CORROSION ALONG **GRAIN BOUNDARIES** BECAUSE OF DIFFERENCE IN COMPOSITION

ERROSION CORROSION

- ATTACK IN METAL DUE TO THE **RELATIVE MOTION** OF A CORROSIVE FLUID AND A METAL SURFACE.

STRESS CORROSION

- COMBINED **ACTION OF AN APPLIED TENSILE STRESS AND A CORROSIVE ENVIRONMENT**

MORE CATHODIC (INERT)	PLATINUM
	GOLD
	GRAPHITE
	TITANIUM
	SILVER
	316 STAINLESS STEEL (PASSIVE)
	NICKEL (PASSIVE)
	COPPER
MORE ANODIC (ACTIVE)	NICKEL (ACTIVE)
	TIN
	LEAD
	316 STAINLESS STEEL (ACTIVE)
	IRON/STEEL
	ALUMINUM ALLOYS
	CADMIUM
	ZINC
	MAGNESIUM

SELECTIVE LEACHING

- FOUND IN SOLID SOLUTION ALLOYS AND OCCURS WHEN ONE ELEMENT IS REMOVED AS A CONSEQUENCE OF CORROSION PROCESSES.
- SELECTIVE LEACHING MAY ALSO OCCUR IN ALUMINUM, IRON, COBALT, CHROMIUM, AND OTHER ELEMENTS

- **DEZINCIFICATION** OF BRASS, IN WHICH ZINC IS SELECTIVELY LEACHED FROM A COPPER-ZINC BRASS ALLOY.

CORROSION PENETRATION RATE

$$CPR = \frac{kW}{\rho A t}$$

CPR - CORROSION PENETRATION RATE
W - WEIGHT LOSS AFTER EXPOSURE
t = TIME
ρ = DENSITY OF MAT.
A = EXPOSED AREA OF SPECIMEN

$k = 534$
CPR IN mils / YEAR (mpy)
▶ 1 mil = 0.001 mm
W IN MILLIGRAMS
ρ IN GRAMS / cm^3
A IN $INCHES^2$
t IN HOURS

$k = 87.6$
CPR IN mm / YEAR (mm/yr)
A IN cm^2

@ CPR OF LESS THAN 20 mpy (0.5 mm/yr) IS ACCEPTABLE!!

INHIBITORS

- ARE SUBSTANCES THAT, WHEN ADDED IN RELATIVELY LOW CONCENTRATIONS TO THE ENVIRONMENT, **DECREASE ITS CORROSIVENESS.**
- **USED IN CLOSED SYSTEMS** SUCH AS AUTOMOBILE RADIATORS AND STEAM BOILERS.

HEAT TREATMENT

- IS AN OPERATION OR COMBINATION OF OPERATIONS INVOLVING THE **HEATING AND COOLING** OF METAL OR AN ALLOY IN THE SOLID STATE FOR THE PURPOSE OF **ALTERING THE PROPERTIES OF MATERIALS.**

TYPES OF HEAT TREATMENT

TYPE	PROCESS	EFFECT
ANNEALING	HEAT **ABOVE** TRANSFORMATION RANGE; **1300-1350 F** SLOW COOLING	- SOFTEN METAL - DECREASE HARDNESS - RELIEVE INTERNAL STRESS
HARDENING	HEAT **ABOVE** TRANSFORMATION TEMPERATURE QUENCHING IN OIL	- INCREASE HARDNESS
NORMALIZING	HEATING **100 F ABOVE** TRANSFORMATION RANGE SUBSEQUENT COOLING AT ROOM TEMPERATURE	- UNIFORM STRUCTURE - RELEASE INTERNAL STRAINS
STRESS RELIEVING	HEATING **BELOW** TRANSFORMATION RANGE, SAY **1100 – 1200 F** HOLDING IT AT THAT TEMPERATURE FOR A SUITABLE TIME	- REDUCE INTERNAL RESIDUAL STRESSES
TEMPERING	**REHEATING** IT BELOW TRANSFORMATION RANGE ANY DESIRED RATE OF COOLING	- ATTAIN DESIRED PROPERTIES - RELIEVE THE INTERNAL STRESSES - **REDUCE BRITTLENESS**
CASE HARDENING	THE SURFACE OR **CASE IS SUBSTANTIALLY HARDER** THAN THE CORE OR INSIDE METAL	- PROVIDE HARD AND WEAR RESISTANT CASE - **RETAIN TOUGHNESS** IN THE CORE

AGING (AND AGE HARDENING) — CHANGE BY WHICH ITS STRUCTURE **RECOVERS FROM AN UNSTABLE CONDITION** PRODUCED BY QUENCHING OR COLD WORKING.

CRITICAL RANGE — HAS THE SAME MEANING AS **TRANSFORMATION RANGE.**

DRAWING — IS OFTEN USED TO MEAN **TEMPERING**

GRAPHITIZING AND ANNEALING PROCESS — CAUSES THE COMBINED CARBON TO **TRANSFORM WHOLLY OR IN PART INTO GRAPHITIC OR FREE CARBON**; IT IS APPLIED TO CAST IRON, SOMETIMES TO HIGH-CARBON STEEL.

MALLEABLIZING — AN ANNEALING PROCESS WHEREBY COMBINED CARBON IN WHITE CAST IRON IS TRANSFORMED WHOLLY OR IN PART TO TEMPER CARBON.

SPHEROIDIZING — IS ANY HEATING AND COOLING OF STEEL THAT PRODUCES A **ROUNDED OR GLOBULAR FORM OF CARBIDE**

TRANSFORMATION RANGE (FOR FERROUS METAL) — TEMPERATURE INTERVAL DURING WHICH **AUSTENITE IS FORMED DURING HEATING**; IT IS ALSO THE TEMPERATURE INTERVAL DURING WHICH **AUSTENITE DISAPPEARS DURING COOLING.**

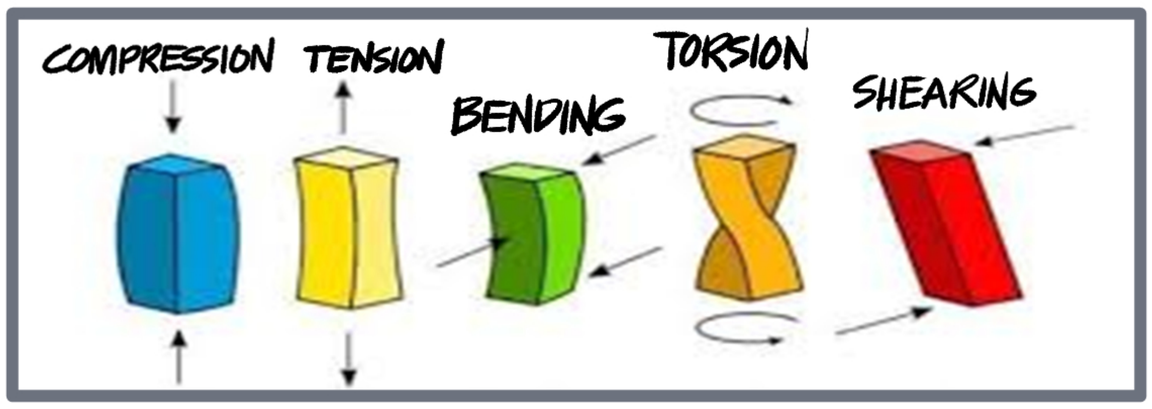
HEAT COLORS FOR STEEL

COLOR	DEG. F
FAINT RED VISIBLE IN DARK	750
FAINT RED	900
BLOOD RED	1050
DARK CHERRY	1075
MEDIUM CHERRY	1250
CHERRY FULL OF RED	1375
BRIGHT RED	1550
SALMON	1650
ORANGE	1725
LEMON	1825
LIGHT YELLOW	1975
WHITE	2200
DAZZLING WHITE	2350

TYPES CASE HARDENING

CARBURIZING	**ADDING CARBON** TO THE SURFACE STEEL BY EXPOSING IT TO HOT CARBONACEOUS SOLIDS, LIQUIDS, OR GASES – **ABOVE THE TRANSFORMATION TEMPERATURE.**
PACK (OR BOX) CARBURIZING	THE PART IS HEATED IN CONTACT WITH **SOLID CARBURIZING COMPOUNDS OF VARIOUS CONSTITUENTS** INCLUDING CHARCOAL, BURNED BONE, CHARRED LEATHER, TAR, AND BARIUM, SODIUM, AND CALCIUM CARBONATES, ESPECIALLY BARIUM CARBONATE AND CHARCOAL.
GAS CARBURIZING	THE PART IS HEATED IN CARBURIZING **GASES** METHANE, ETHANE, PROPANE, AND CO.
LIQUID CARBURIZING	THE PART IS IMMERSED IN **A MOLTEN SALT BATH**
CYANIDING	IS ACCOMPLISHED BY IMMERSING THE PART IN A HOT (ABOUT **1550 F**) LIQUID SALT BATH, **SODIUM CYANIDE** BEING A COMMON MEDIUM IN BOTH PROCESSES.
CARBONITRIDING	SIMULTANEOUS **ABSORPTION OF CARBON AND NITROGEN** FROM A SURROUNDING HOT GASEOUS ATMOSPHERE, FOLLOWED BY EITHER QUENCHING OR SLOW COOLING
INDUCTION HARDENING	CONSIST OF HEATING A THIN SURFACE LAYER, PREFERABLY OF ANNEALED OR NORMALIZED STEEL, ABOVE THE TRANSFORMATION RANGE **BY ELECTRICAL INDUCTION**
FLAME HARDENING	**HEATING THE SURFACE OF AN IRON-BASE ALLOY**, WHICH IS PREFERABLY ANNEALED OR NORMALIZED, AND THEN QUENCHING IT.

STRENGTH OF MATERIALS



TYPES OF SIMPLE LOADING

TENSION | COMPRESSION

$$S = \frac{F}{A}$$

FOR SOLID CIRCULAR: $A = \pi/4\, D^2$

FOR HOLLOW: $A = \frac{\pi}{4}(D_o^2 - D_i^2)$

FOR DUCTILE MAT'LS:

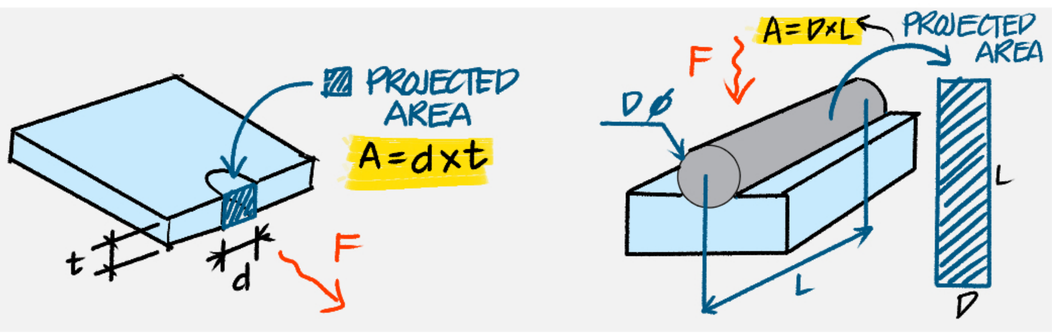
@ NO DEFORMATION $S_d = \frac{S_y}{N_{sf}}$ — S_y = YIELD STRENGTH

@ DEFORMATION $S_d = \frac{S_u}{N_{sf}}$ — S_u = ULTIMATE S.

FOR BRITTLE: $S_d = \frac{S_u}{N_{sf}}$ — S_u = ULT. STR.

BEARING

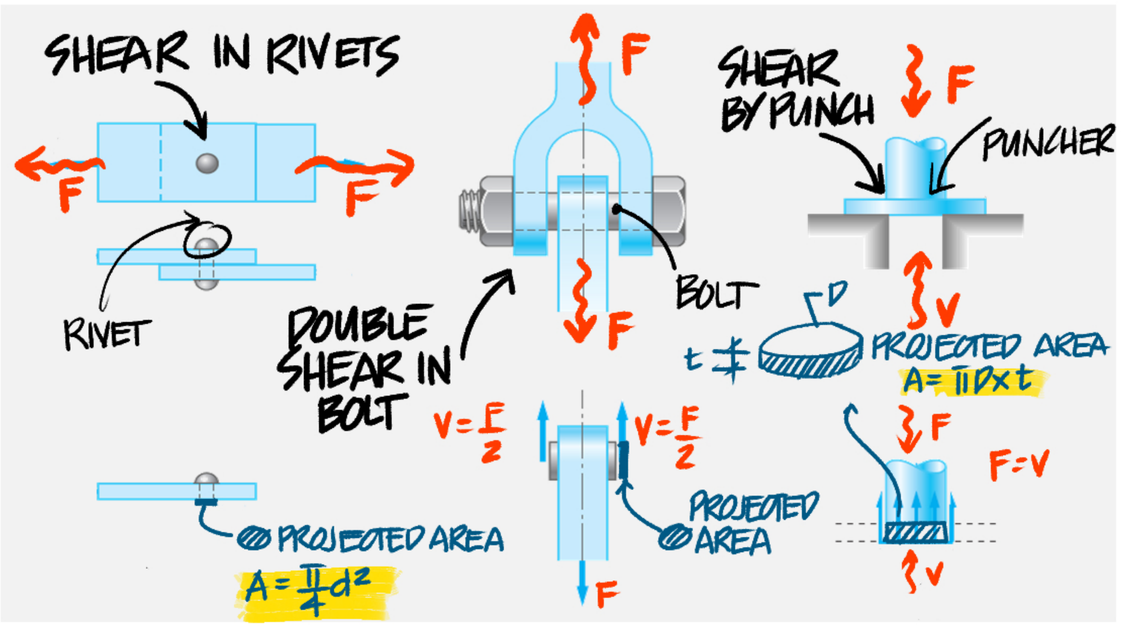
CONTACT PRESSURE BETWEEN THE SEPARATE BODIES. IT DIFFERS FROM COMPRESSIVE STRESS, AS IT IS AN INTERNAL STRESS CAUSED BY COMPRESSIVE FORCES.



$$S_b = F / \text{PROJECTED AREA}$$

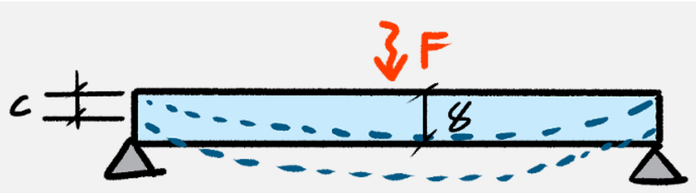
SHEARING (TANGENTIAL STRESS)

CAUSE DEFORMATION OF A MATERIAL BY SLIPPAGE ALONG A PLANE OR PLANES PARALLEL TO THE IMPOSED STRESS.



$$S_s = F / \text{PROJECTED AREA}$$

BENDING | FLEXURAL | NORMAL



$$\delta = \frac{FL}{AE}$$

$$S_f = \frac{Mc}{I} \qquad S_f = \frac{M}{Z} \qquad Z = \frac{I}{c}$$

M = MAX. BENDING MOMENT
I = RECT. MOMENT OF INERTIA
Z = SECTION MODULUS

SOLID CIRCULAR

$c = \frac{D}{2}$ $\quad I = \frac{\pi}{64} D^4$

$$S_f = \frac{32M}{\pi D^3}$$

HOLLOW CIRC.

$c = \frac{D_o}{2}$ $\quad I = \frac{\pi}{64}(D_o^4 - D_i^4)$

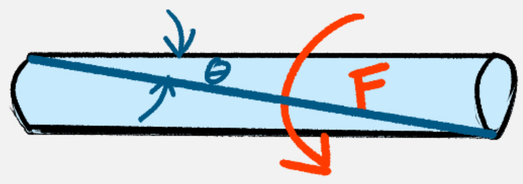
$B = \frac{D_i}{D_o}$

$$S_f = \frac{32 M D_o}{\pi(D_o^4 - D_i^4)} \qquad S_f = \frac{32M}{\pi D_o^3 (1 - B^4)}$$

RECT.

$$I = \frac{bh^3}{12} \qquad c = \frac{h}{2}$$

TORSION (DUE TO TWISTING)



$$\theta = \frac{TL}{JG}$$

$$S_{st} = \frac{Tc}{J} \qquad S_{st} = \frac{T}{Z} \qquad Z = \frac{J}{c}$$

T = MAXIMUM TORQUE
J = POLAR MOMENT OF INERTIA
Z = POLAR SECTION MODULUS

SOLID CIRCULAR

$c = \frac{D}{2}$ $\quad J = \frac{\pi}{32} D^4$

$$S_f = \frac{16T}{\pi D^3}$$

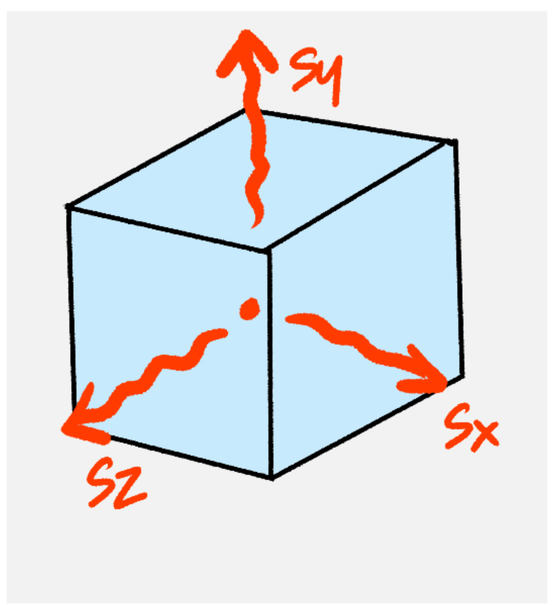
HOLLOW CIRC.

$c = \frac{D_o}{2}$ $\quad J = \frac{\pi}{32}(D_o^4 - D_i^4)$

$B = \frac{D_i}{D_o}$

$$S_f = \frac{16T D_o}{\pi(D_o^4 - D_i^4)} \qquad S_f = \frac{16T}{\pi D_o^3 (1 - B^4)}$$

TRI-AXIAL STRESS



ε = DIRECT + INDIRECT

$$\varepsilon_x = \frac{S_x}{E} - M\left(\frac{S_y}{E} + \frac{S_z}{E}\right)$$

$$\varepsilon_y = \frac{S_y}{E} - M\left(\frac{S_x}{E} + \frac{S_z}{E}\right)$$

$$\varepsilon_z = \frac{S_z}{E} - M\left(\frac{S_x}{E} + \frac{S_y}{E}\right)$$

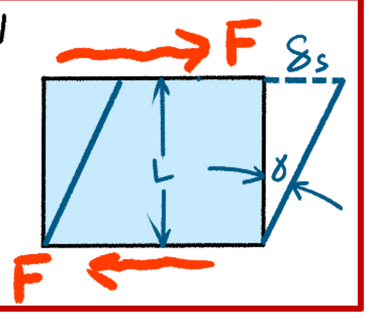
ε = STRAIN M = POISSON'S :
E = MODULUS OF ELASTICITY

STRAIN | ELONGATION

$S \propto \varepsilon$
$S = E\varepsilon$
$\varepsilon = \frac{\Delta L}{L} = \frac{\delta}{L}$
$\delta = \frac{FL}{AE}$

APPLYING HOOKE'S LAW TO SHEAR:

$S_s \propto \varepsilon$
$S = G\varepsilon$
$\delta_s = \frac{FL}{AG}$



TOTAL STRAIN ENERGY

ENERGY STORED IN A LOADED MEMBER IS EQUAL TO THE WORK REQUIRED TO DEFORM THE MEMBER.

$$U = \frac{1}{2}F(\Delta L) = \frac{1}{2}\frac{F^2 L}{AE} \quad | \quad \Delta L = \delta$$

SHEAR DEFORMATION

$$S_s = G\varepsilon_s \qquad \varepsilon_s = \frac{\delta_s}{L} \rightarrow \delta_s = \frac{FL}{AG}$$

POISSON'S RATIO (M)

$$M = -\frac{\text{LATERAL } \varepsilon}{\text{LONGITUDINAL } \varepsilon} \quad | \quad M = \frac{E}{2G} - 1$$

BULK MODULUS (K)

A MEASURE OF HOW RESISTANT TO COMPRESSION THAT SUBSTANCE IS.

$$K = \frac{S_{VOL}}{\varepsilon_{VOL}} \rightsquigarrow S_{VOL} = \text{VOL. STRESS}, \quad \varepsilon_{VOL} = \text{VOL. STRAIN} = \frac{\Delta V}{V} \quad | \quad K = \frac{E}{3(1-2M)}$$

CONSTRAINED MODULUS (D)

$$D = \frac{E \times (1-M)}{(1-2M)(1+M)}$$

THERMAL ELONGATION

$\Delta L = \alpha L \Delta t$

$\Delta A = \gamma A \Delta t$

$\Delta \text{Vol.} = \beta V \Delta t$

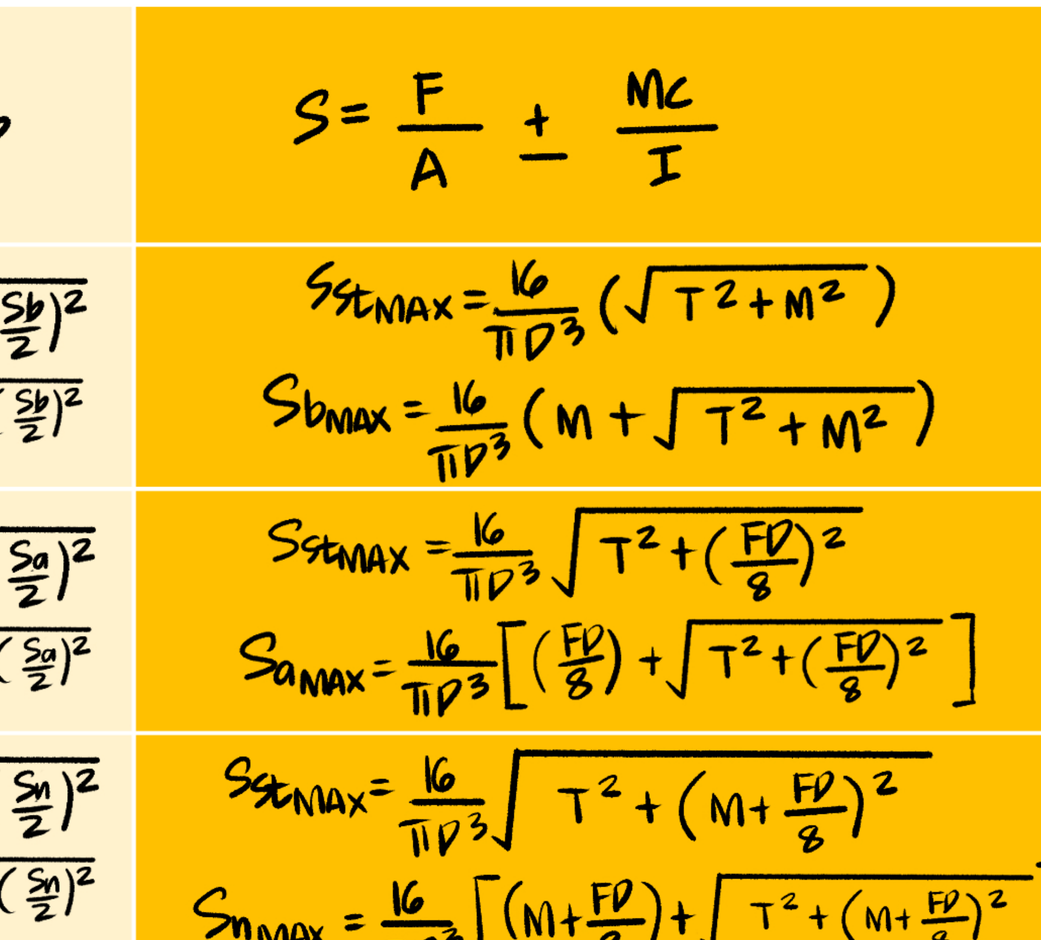
$\varepsilon = \text{STRAIN} = \Delta L / L$

$\Delta L = \delta$

$\varepsilon = \alpha \Delta t$ | $S = \alpha E \Delta t$

COMBINED STRESS

MOHR'S CIRCLE



COMBINED STRESS	GENERAL FORMULA	SHORTCUT
AXIAL AND BENDING	$S = S_a \pm S_b$	$S = \frac{F}{A} \pm \frac{Mc}{I}$
BENDING AND TORSION	$S_{st\,MAX} = \sqrt{S_{st}^2 + \left(\frac{S_b}{2}\right)^2}$ $S_{b\,MAX} = \frac{S_b}{2} + \sqrt{S_{st}^2 + \left(\frac{S_b}{2}\right)^2}$	$S_{stMAX} = \frac{16}{\pi D^3}\left(\sqrt{T^2 + M^2}\right)$ $S_{bMAX} = \frac{16}{\pi D^3}\left(M + \sqrt{T^2 + M^2}\right)$
AXIAL AND TORSION	$S_{st\,MAX} = \sqrt{S_{st}^2 + \left(\frac{S_a}{2}\right)^2}$ $S_{a\,MAX} = \frac{S_a}{2} + \sqrt{S_{st}^2 + \left(\frac{S_a}{2}\right)^2}$	$S_{stMAX} = \frac{16}{\pi D^3}\sqrt{T^2 + \left(\frac{FD}{8}\right)^2}$ $S_{aMAX} = \frac{16}{\pi D^3}\left[\left(\frac{FD}{8}\right) + \sqrt{T^2 + \left(\frac{FD}{8}\right)^2}\right]$
AXIAL, BENDING AND TORSION	$S_{st\,MAX} = \sqrt{S_{st}^2 + \left(\frac{S_n}{2}\right)^2}$ $S_{n\,MAX} = \frac{S_n}{2} + \sqrt{S_{st}^2 + \left(\frac{S_n}{2}\right)^2}$ $S_n = S_a + S_b$	$S_{stMAX} = \frac{16}{\pi D^3}\sqrt{T^2 + \left(M + \frac{FD}{8}\right)^2}$ $S_{nMAX} = \frac{16}{\pi D^3}\left[\left(M + \frac{FD}{8}\right) + \sqrt{T^2 + \left(M + \frac{FD}{8}\right)^2}\right]$

BUCKLING

REFERS TO THE **LOSS OF STABILITY** OF A COMPONENT AND IS USUALLY INDEPENDENT OF MATERIAL STRENGTH.

RADIUS OF GYRATION

IS THAT DISTANCE FROM THE AXIS AT WHICH THE ENTIRE MASS OF A BODY MAY BE CONSIDERED AS CONCENTRATED, THE MOMENT OF INERTIA, MEANWHILE, REMAINING UNCHANGED.

IF AXIAL LOAD IS **COMPRESSIVE**:

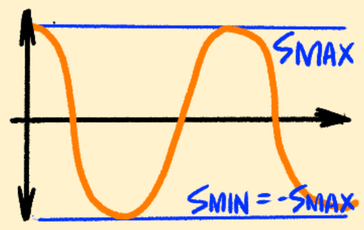
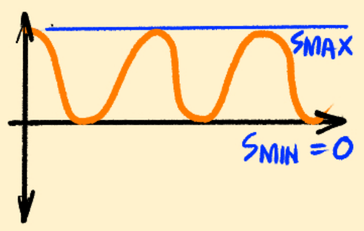
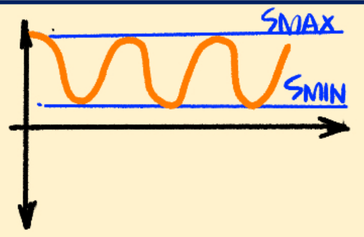
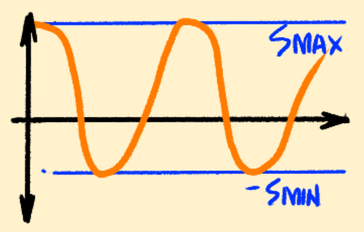
→ MULTIPLY $\left(\frac{FD}{8}\right) \times B$ ← BUCKLING FACTOR

$$B = \frac{1}{1 - 0.0044\left(\frac{L}{K}\right)}$$

K = LEAST RADIUS OF GYRATION

- FOR SOLID SHAFT: $K = D/4$

VARIABLE STRESS

LOAD	PROFILE	MEAN STRESS	ALTERNATING STRESS
REVERSED		0	S_{MAX}
REPEATED		$\frac{S_{MAX}}{2}$	$\frac{S_{MAX}}{2}$
FLUCTUATING		$\frac{1}{2}(S_{MAX}+S_{MIN})$	$\frac{1}{2}(S_{MAX}-S_{MIN})$
ALTERNATING		$\frac{1}{2}(S_{MAX}-S_{MIN})$	$\frac{1}{2}(S_{MAX}+S_{MIN})$

GENERAL FORMULA

MEAN STRESS:

$$S_m = \frac{1}{2}(S_{MAX} + S_{MIN})$$

ALTERNATING STRESS:

$$S_a = \frac{1}{2}(S_{MAX} - S_{MIN})$$

SODERBERG'S EQ.

FOR DUCTILE MAT'LS:

$$\frac{1}{N_{sf}} = \frac{S_m}{S_y} + \frac{S_a}{S_n}$$

S_y → YIELD

GOODMAN'S EQ.

FOR BRITTLE MAT'LS:

$$\frac{1}{N_{sf}} = \frac{S_m}{S_u} + \frac{S_a}{S_n}$$

S_u → ULTIMATE

STRESS CONCENTRATION

IS THE CONDITION WHICH CAUSES **ACTUAL STRESSES IN MACHINE MEMBERS TO BE HIGHER THAN NOMINAL VALUES** PREDICTED BY THE ELEMENTARY DIRECT AND COMBINED STRESS EQUATION.

THANKS G!

THREADS, **H**OLES, **A**BRUPT CHANGES IN CROSS SECTION, **N**OTCHES, **K**EYWAYS, **S**PLINES, **G**ROOVES

STRESS CONCENTRATION FACTOR (K)

$$k = \frac{S_{MAX}}{S_N}$$

S_{MAX} = MAX. STRESS DUE TO S. RAISERS
S_N = NOMINAL STRESS OR COMBINED S.

NOTCH SENSITIVITY

IS THE CONDITION WHICH CAUSES **ACTUAL STRESS IN A MACHINE MEMBER UNDER FATIGUE CONDITIONS**

NOTCH SENSITIVITY FACTOR (q)

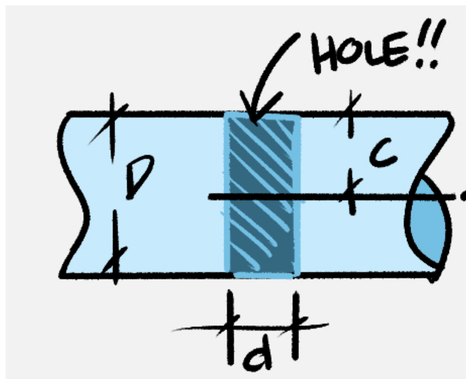
IS AN INDICATION OF THE DEGREE OF THEORETICAL STRESS CONCENTRATION OBTAINED UNDER **FATIGUE CONDITION**

$$q = \frac{K_f - 1}{K_t - 1}$$

K_f = STRESS CON. FACTOR FOR FATIGUE LOAD
K_t = STRESS CON. FACTOR FOR STATIC LOAD

$$q = \frac{1}{1 + \frac{a}{r}}$$

r = RADIUS OF CURVATURE
a = 0.0025 (Q&T STEELS)
a = 0.01 (A&N STEELS)



PROP.	STEEL	COPPER	ALU
E (SI)	210 GPa	110	70
E (ENG)	30×10^6 psi	16×10^6	10
G (SI)	83 GPa	46	27
G (ENG)	12×10^6 psi	6.7×10^6	3.9
M	0.28	0.34	0.33
α (SI)	11.7×10^{-6} °C^{-1}	17×10^{-6}	23×10^{-6}
α (ENG)	6.5×10^{-6} °F^{-1}	9.4×10^{-6}	12.8×10^{-6}
ρ (SI)	7850 $\frac{kg}{m^3}$	8940	2770
ρ (ENG)	490 $\frac{lb}{ft^3}$	559	173

FOR SHAFT WITH RADIAL HOLE

FOR BENDING

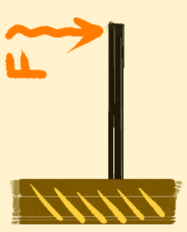
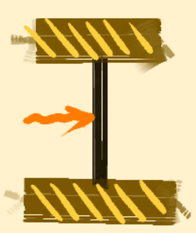
$$S = \frac{M}{\frac{\pi D^3}{32} - \frac{dD^2}{6}}$$

FOR TORSION

$$S = \frac{T}{\frac{\pi D^3}{16} - \frac{dD^2}{6}}$$

BEAMS

SIMPLIFIED →

ANGULAR DEFLEC.	VARIABLE V	$\frac{wL^3}{24EI}$	$\frac{7wL^3}{360EI}$	—	—
	DISTRIBUTED D	$\frac{wL^3}{6EI}$	$\frac{wL^3}{24EI}$	NO ANGULAR DEFLECTION!	—
	UNIFORM U	$\frac{FL^2}{2EI}$	$\frac{FL^2}{16EI}$	—	—
DEFLECTION	VARIABLE V	$\frac{wL^4}{30EI}$	$\frac{2.5wL^4}{384EI}$	$\frac{wL^4}{768EI}$	—
	DISTRIBUTED D	$\frac{wL^4}{8EI}$	$\frac{5wL^4}{384EI}$	$\frac{wL^4}{384EI}$	—
	UNIFORM U	$\frac{FL^3}{3EI}$	$\frac{FL^3}{48EI}$	$\frac{FL^3}{192EI}$	—
MOMENT	VARIABLE V	$\frac{wL^2}{6}$	$\frac{wL^2}{9\sqrt{3}}$	@A $\frac{wL^2}{30}$ @B $\frac{wL^2}{20}$	—
	DISTRIBUTED D	$\frac{wL^2}{2}$	$\frac{wL^2}{8}$	$\frac{wL^2}{12}$	—
	UNIFORM U	FL	FL/4	FL/8	—
		CANTILEVER C 	SIMPLY SUPPORTED S	FIXED F 	

REMEMBER THIS!

MOMENT

	U	D	V
C	1	1/2	1/6
S	1/4	1/8	$\frac{1}{9\sqrt{3}}$
F	1/8	1/12	1/30 / 1/20

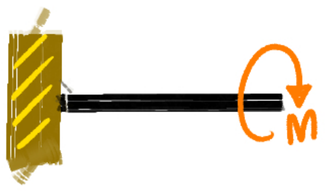
DEFLECTION

	U	D	V
C	1/3	1/8	1/30
S	8/384	5/384	2.5/384
F	2/384	1/384	0.5/384

θ DEFLECTION

	U	D	V
C	1/2	1/6	1/24
S	22.5/360	15/360	7/360
F	—	—	—

CANTILEVER BEAM W/ END MOMENT



$M = M$

$\delta = \frac{ML^2}{2EI}$

$\theta = ML/EI$

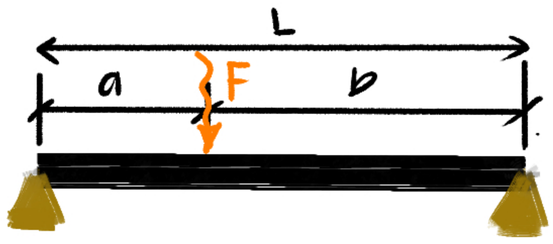
CANTILEVER BEAM @ MIDSPAN

$M = \frac{FL}{2}$

$\delta = \frac{40FL^3}{384EI}$

$\theta = \frac{FL^2}{8EI}$

SIMPLY SUPPORTED BEAM W/ INTERMEDIATE LOAD



$M = \left(\frac{ab}{L}\right)(F)$

$\delta = \frac{b(L^2-b^2)^{3/2}}{9\sqrt{3}\,L}\left(\frac{F}{EI}\right)$

$\theta_1 = \frac{Pab(1+b/L)}{6EI}$ $\theta_2 = \frac{Pab(1+a/L)}{6EI}$

RECTANGULAR BEAMS

S_f = FLEXURAL STRESS

$S_f = \frac{6M}{bh^2}$ → MAXIMUM MOMENT

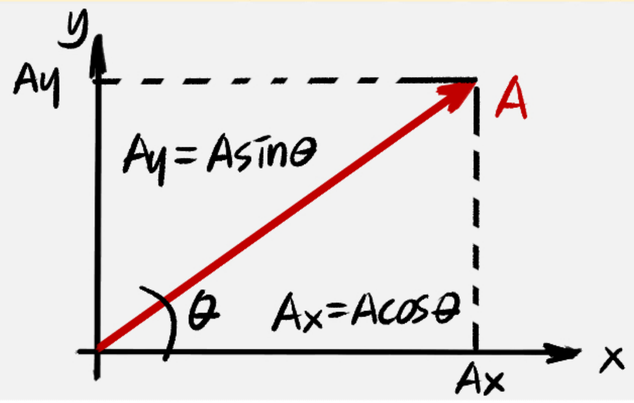
S_s = SHEARING STRESS

$S_s = \frac{3V}{2bh}$ → MAXIMUM SHEAR

ENGINEERING MECHANICS

TWO GENERAL CLASSES OF SHAFTS	
STATICS	FORCES AND EFFECTS ON **BODIES AT REST**
DYNAMICS	MOTION AND EFFECTS OF FORCES ON **MOVING BODIES**
KINEMATICS	MOTION WITHOUT FORCES
KINETICS	FORCES AND THEIR EFFECT IN ACCELERATION

VECTOR REPRESENTATION



☑ RECTANGULAR FORM

$A = Ax\,\bar{i} + Ay\,j$

☑ POLAR FORM

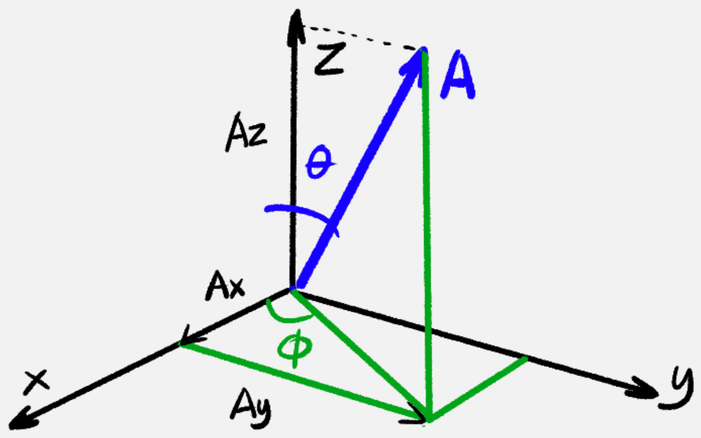
$A = |A| \angle \theta$

$|A| = \sqrt{(Ax)^2 + (Ay)^2}$

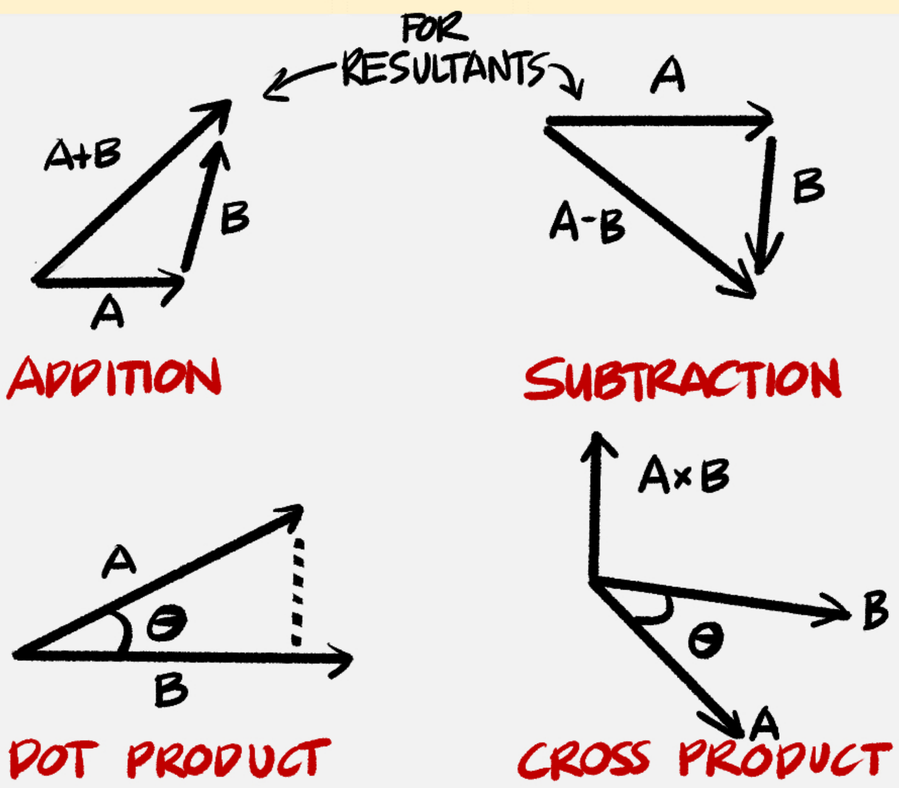
$\theta = \tan^{-1}\left(\frac{Ay}{Ax}\right)$

STATICS

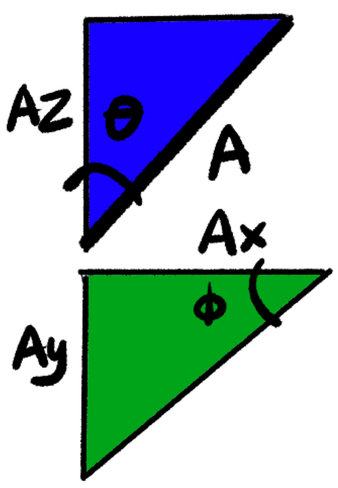
VECTOR REPRESENTATION (3D)



VECTOR OPERATIONS



ANALYSIS



☑ RESULTANT:

$$A = \sqrt{(Ax)^2 + (Ay)^2 + (Az)^2}$$

$$\cos\theta = \frac{Az}{\sqrt{(Ax)^2 + (Ay)^2 + (Ay)^2}}$$

$$\tan\phi = \frac{Ay}{Ax}$$

DOT PRODUCT

➤ FOR WORK

$\vec{A} \cdot \vec{B} = |A||B|\cos\theta$

$\vec{A} \cdot \vec{B} = Ax Bx + Ay By + Az + Bz$

$|A| = \sqrt{Ax^2 + Ay^2 + Az^2}$

$|B| = \sqrt{Bx^2 + By^2 + Bz^2}$

$\cos\theta = \frac{\vec{A} \cdot \vec{B}}{|A||B|}$

IF $\theta = 90°$, $\cos\theta = 0$

DOT PRODUCT = 0

VECTORS PERPENDICULAR

CROSS PRODUCT

➤ FOR TORQUE

$\vec{A} \times \vec{B} = |A||B| \sin\theta$

EXAMPLE:

$A = 1\bar{i} - 2j + 3K$

$B = 3i + 4j - 5K$

$$\begin{bmatrix} \bar{i} & j & K \\ 1 & -2 & 3 \\ 3 & 4 & -5 \end{bmatrix}$$

$$\vec{A} \times \vec{B} = i\begin{vmatrix} -2 & 3 \\ 4 & -5 \end{vmatrix} - j\begin{vmatrix} 1 & 3 \\ 3 & -5 \end{vmatrix} + K\begin{vmatrix} 1 & -2 \\ 3 & 4 \end{vmatrix}$$

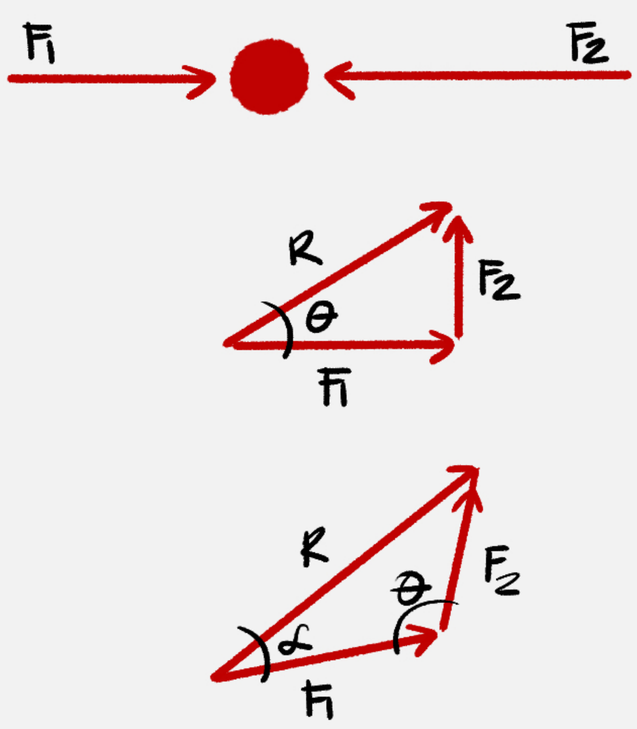
$\sin\theta = \frac{\vec{A} \times \vec{B}}{|A||B|}$

IF $\theta = 0°$, $\sin\theta = 0$

CROSS PRODUCT = 0

VECTORS ARE PARALLEL

RESULTANT OF 2 OR MORE CONCURRENT FORCES



COLINEAR FORCES:

$R = \Sigma F = F_1 - F_2$

FORCES FORMING A RIGHT △

$R = \sqrt{F_1^2 + F_2^2}$ $\quad \tan\theta = \frac{F_2}{F_1}$

USING COSINE LAW:

$R^2 = F_1^2 + F_2^2 - 2(F_1)(F_2)\cos\theta$

USING SINE LAW:

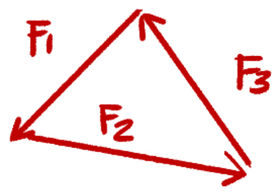
$\frac{R}{\sin\theta} = \frac{F_2}{\sin\alpha}$

STATIC EQUILIBRIUM

↳ STATE OF THE BODY @ REST

POLYGON CONDITION

- RESULTANT IS ZERO!



COMPONENT CONDITION

$\Sigma F_x = 0$; $\Sigma F_y = 0$; $\Sigma F_z = 0$

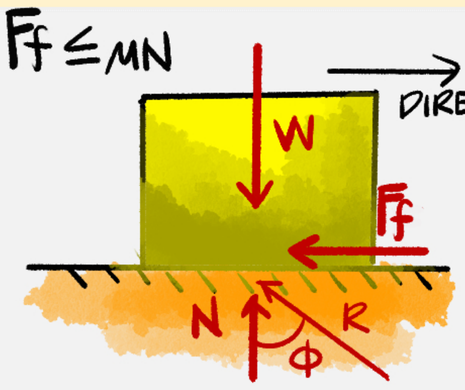
FRICTION

STATIC FRICTION – STATIONARY SURFACES

DYNAMIC FRICTION – RELATIVE (DYNAMIC) MOTION

ROLLING FRICTION – SURFACE ROTATION

SLIDING BLOCK



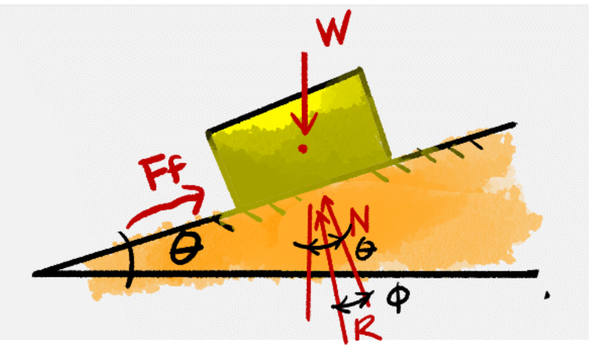
N = NORMAL REACTION

$\mu = \tan\phi$

ϕ = ∠ OF FRICTION

NOTE:

- ⊙ $F_f = \mu N$ → ABOUT TO MOVE FROM REST
- ⊙ $F_f < \mu N$ → OTHERWISE

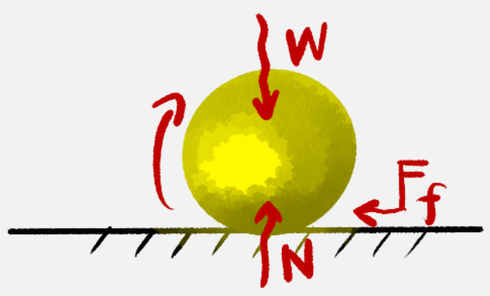


MAXIMUM INCLINATION OF AN INCLINED PLANE W/O SLIDING

W/O SLIDING:

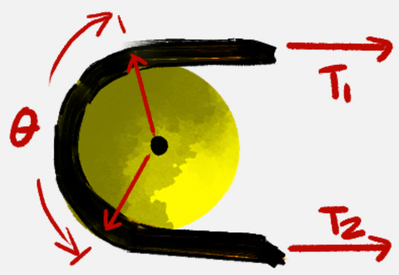
$$\phi = \theta$$

FRICTION ON ROLLING OBJECTS



$$F_f = \mu\left(\frac{W}{r}\right)$$

BELT FRICTION



$$\frac{T_1}{T_2} = e^{f\theta}$$

f = COEFF. OF FRICTION

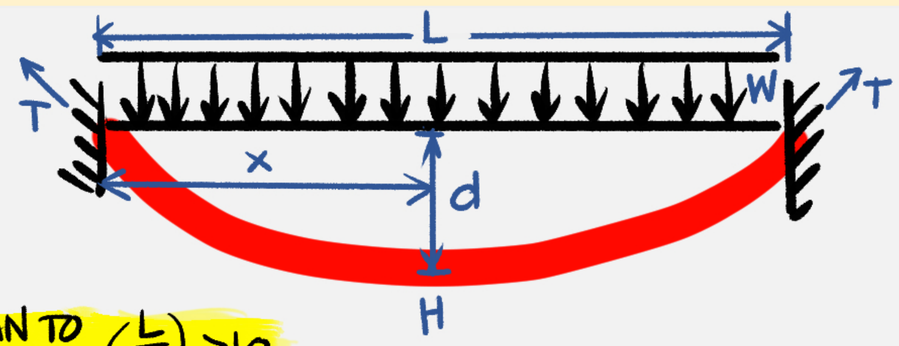
θ = ∠ OF CONTACT, rad

PARABOLIC CABLES

SHAPE OF SUSPENDED CABLE IS *ASSUMED TO BE A PARABOLA*

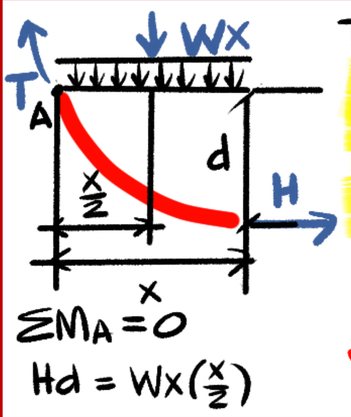
- DISTRIBUTED ALONG THE *SPAN* OF THE CABLE

SYMMETRICAL SUPPORTS



SPAN TO SAG RATIO $\left(\frac{L}{d}\right) > 10$

ANALYSIS



$\Sigma M_A = 0$

$Hd = Wx\left(\frac{x}{2}\right)$

THUS;

$$H = \frac{Wx^2}{2d}$$

$$T = \sqrt{(Wx)^2 + H^2}$$

LENGTH;

$$S = L + \frac{8d^2}{3L} - \frac{32d^4}{5L^3}$$

USE LENGTH OF ARC FORMULA:

$$S = \int\sqrt{(y')^2 + 1}\ dx$$

CATENARY → HYPERBOLIC

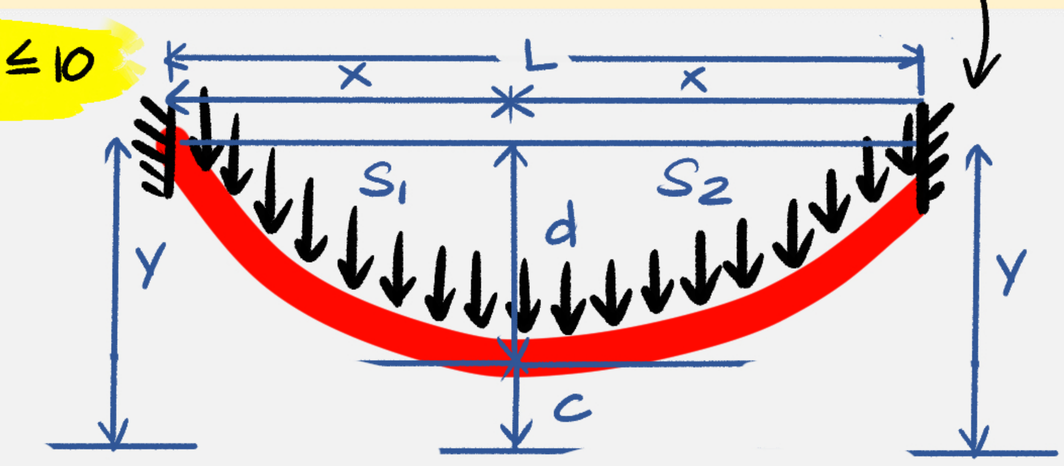
- DISTRIBUTED ALONG THE *LENGTH* OF THE CABLE

$y = \cosh x$

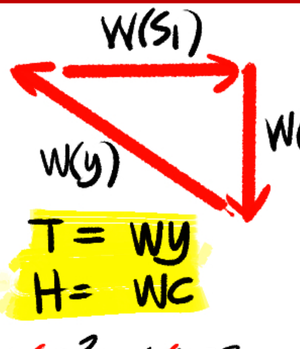
SYMMETRICAL SUPPORTS

GRAPH

$\frac{L}{d} \leq 10$



ANALYSIS



$T = Wy$

$H = Wc$

$(Wc)^2 + (WS_1)^2 = (Wy)^2$

$$c^2 + s^2 = y^2$$

$$y = c\cosh\left(\frac{x}{c}\right)$$

$$S = c\sinh\left(\frac{x}{c}\right)$$

FOR SYMMETRICAL

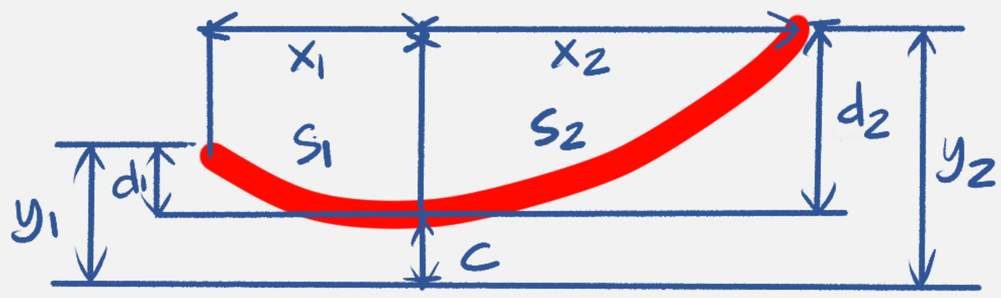
$S = S_1 = S_2$

$$x = c\ln\left(\frac{S+y}{c}\right)$$

SPAN = 2x

TOTAL LENGTH = $S_1 + S_2$

UNSYMMETRICAL SUPPORTS



ANALYSIS

$$T_1 = wy_1 \qquad T_2 = wy_2$$

$$y_1^2 = c^2 + s_1^2 \qquad y_2^2 = c^2 + s_2^2$$

$$x_1 = c\ln\left(\frac{S_1 + y_1}{c}\right) \qquad x_2 = c\ln\left(\frac{S_2 + y_2}{c}\right)$$

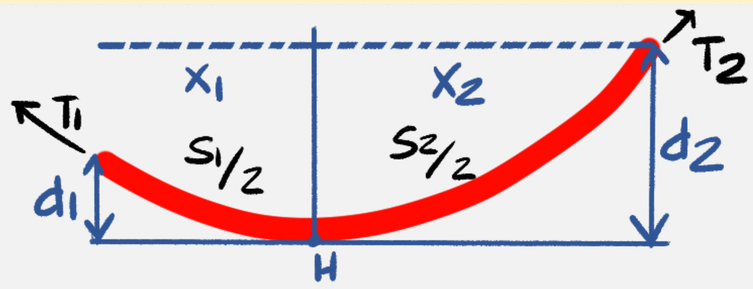
$$Y_1 = c\cosh\left(\frac{x_1}{c}\right) \qquad Y_2 = c\cosh\left(\frac{x_2}{c}\right)$$

$$S_1 = c\sinh\left(\frac{x_1}{c}\right) \qquad S_2 = c\sinh\left(\frac{x_2}{c}\right)$$

SPAN = $x_1 + x_2$

TOTAL LENGTH = $S_1 + S_2$

UNSYMMETRICAL SUPPORTS



ANALYSIS

$$H = \frac{Wx_1^2}{2d_1} = \frac{Wx_2^2}{2d_2}$$

FOR TENSION:

$$T_1 = \sqrt{(Wx_1)^2 + H^2}$$

$$T_2 = \sqrt{(Wx_2)^2 + H^2}$$

$$\text{TOTAL LENGTH} = \frac{S_1 + S_2}{2}$$

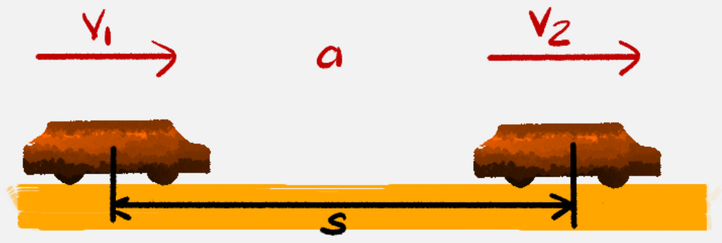
$2x = L$ = SPAN

d = SAG

S = TOTAL LENGTH

DYNAMICS

RECTILINEAR MOTION



ANALYSIS

CASE I : CONSTANT VELOCITY

$s = vt$

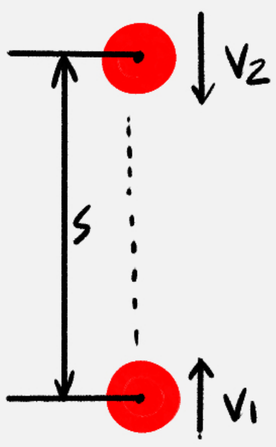
$a = 0$

CASE II : CONSTANT ACCELERATION

- $V_2 = V_1 \pm at$
- $V_2^2 = V_1^2 \pm 2as$
- $s = V_1 t \pm \frac{1}{2}at^2$

NOTE:
+ ACCELERATION
− DECELERATION

FREE-FALLING BODY



ANALYSIS

- $V_2 = V_1 \pm gt$
- $V_2^2 = V_1^2 \pm 2gs$
- $s = V_1 t \pm \frac{1}{2}gt^2$

GOING UP (−) ↑
GOING DOWN (+) ↓

MAXIMUM HEIGHT ; $V_2 = 0$

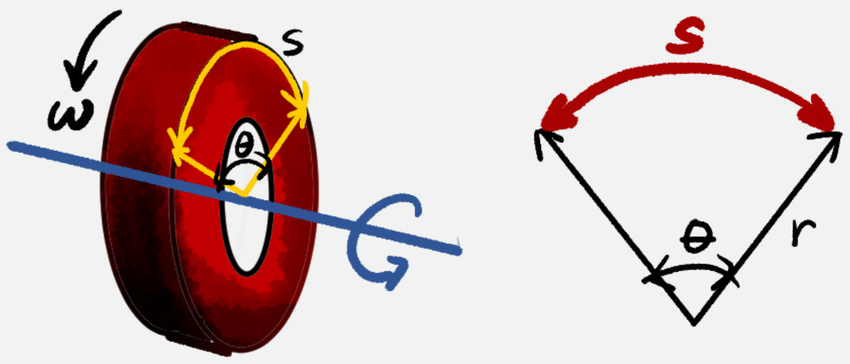
$s = \frac{V_1^2}{2g}$

$t = \frac{V_1}{g}$

TOTAL TIME OF FLIGHT

$t = \frac{2V_1}{g}$

ROTATIONAL KINEMATICS



ANALYSIS

CASE I : CONSTANT VELOCITY

$\theta = \omega t$

CASE II : CONSTANT ACCELERATION

- $\omega_2 = \omega_1 \pm \alpha t$
- $\omega_2^2 = \omega_1^2 \pm 2\alpha\theta$
- $\theta = \omega_1 t \pm \frac{1}{2}\alpha t^2$

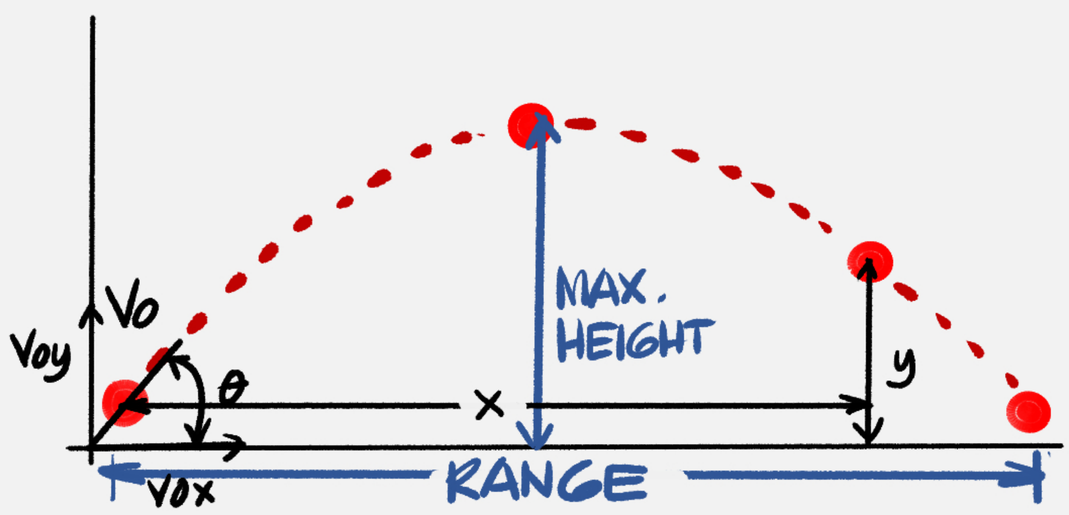
θ = ∠ DISPLACE. ; rad
ω = ∠ VEL ; $\frac{rad}{s}$
α = ∠ ACCEL ; $\frac{rad}{s^2}$
t = TIME

RELATIONSHIP BETWEEN TRANSLATION AND ROTATOIN

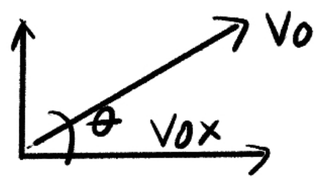
$s = r\theta$
$V = r\omega$
$a = r\alpha$

θ = IN RADIANS
ω = IN RAD/SEC
α = IN RAD/SEC^2

PROJECTILE



X COMPONENT (CONSTANT VELOCITY) $a_x = 0$



$V_{ox} = V_o \cos\theta$; $x = V_{ox} t$

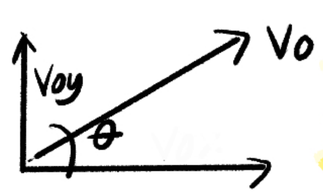
$V_x = V_{ox}$

Y COMPONENT (FREE-FALLING BODIES) $a_y = -g$

$V_y = V_{oy} - gt$

$V_y^2 = V_{oy}^2 - 2gY$

$Y = V_{oy} t - \frac{1}{2}gt^2$



$V_{oy} = V_o \sin\theta$

$y = (V_o \sin\theta)t - \frac{1}{2}gt^2$

RANGE AND SUMMIT

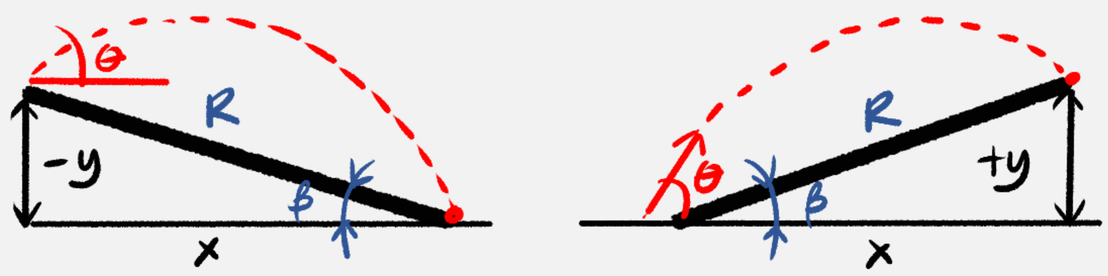
MAXIMUM HEIGHT

$h = \frac{V_o^2 \sin^2\theta}{2g}$

MAXIMUM RANGE

$R = \frac{V_o^2 \sin 2\theta}{g}$

GENERAL FORMULA

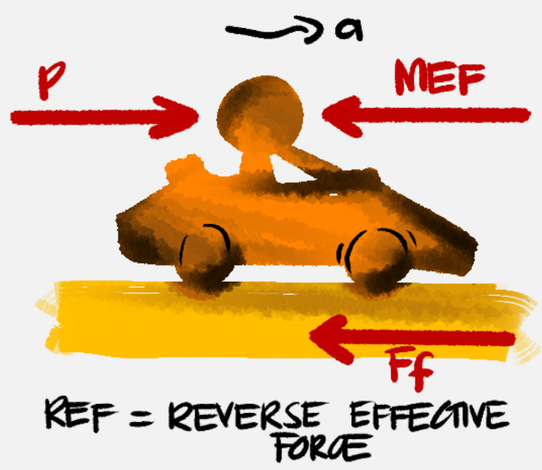


ANALYSIS

$$\pm y = x\tan\theta - \frac{gx^2}{2V_o^2\cos^2\theta}$$

$R^2 = x^2 + y^2 \rightarrow \sin\beta = \frac{y}{R}$; $\cos\beta = \frac{x}{R}$

KINETICS



REF = REVERSE EFFECTIVE FORCE

NEWTON'S LAW

2ND LAW:

$F = ma$

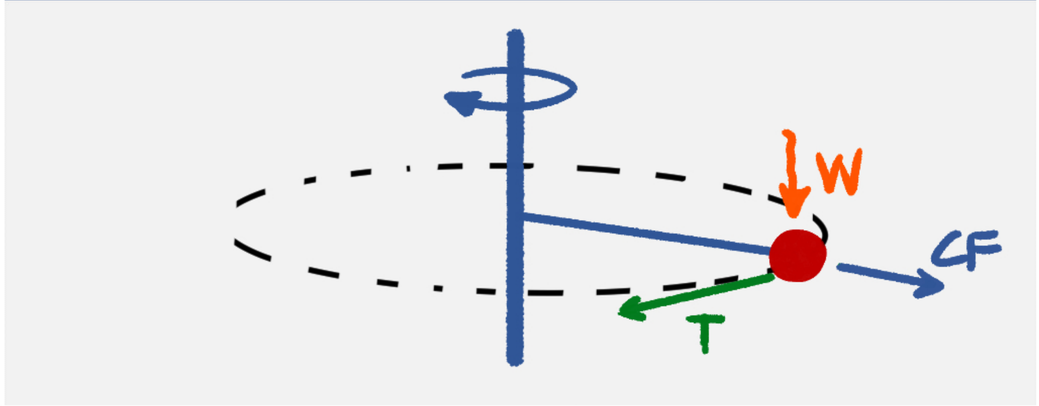
D'ALEMBERT'S PRINCIPLE

3RD LAW ↗

$REF = P - F_f$

$REF = \frac{W}{g}a = ma$

CENTRIFUGAL FORCE



CENTRIFUGAL FORCE – PSEUDO / MISNOMER FORCE
↳ RATHER, ABSENCE OF INWARD POINTING CENTRIPETAL FORCE

DIRECTED **AWAY** FROM THE AXIS OF ROTATION

– AWAY FROM CENTER

$$CF = m\omega^2 r = m\left(\frac{V}{r}\right)^2 r = \frac{W}{g} \times \frac{v^2}{r}$$

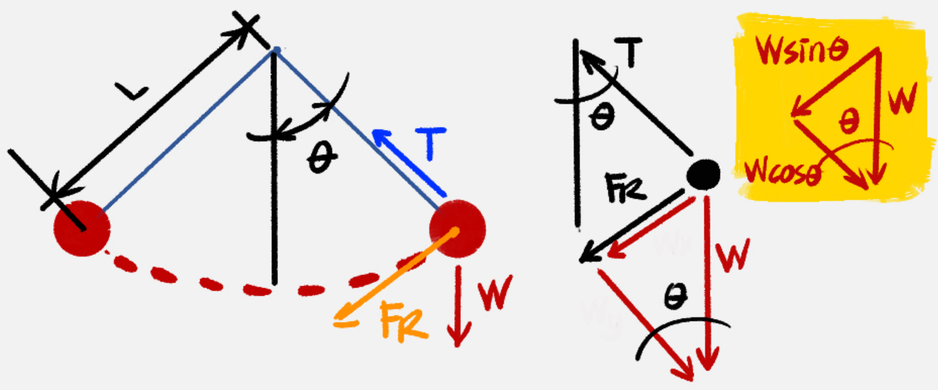
CENTRIPETAL FORCE (TANGENTIAL INERTIA FORCE)

ALSO KNOWN AS REVERSED TANGENTIAL EFFECTIVE FORCE

– TOWARDS THE CENTER

$$T = ma = \frac{W}{g}(r\alpha)$$

SIMPLE PENDULUM



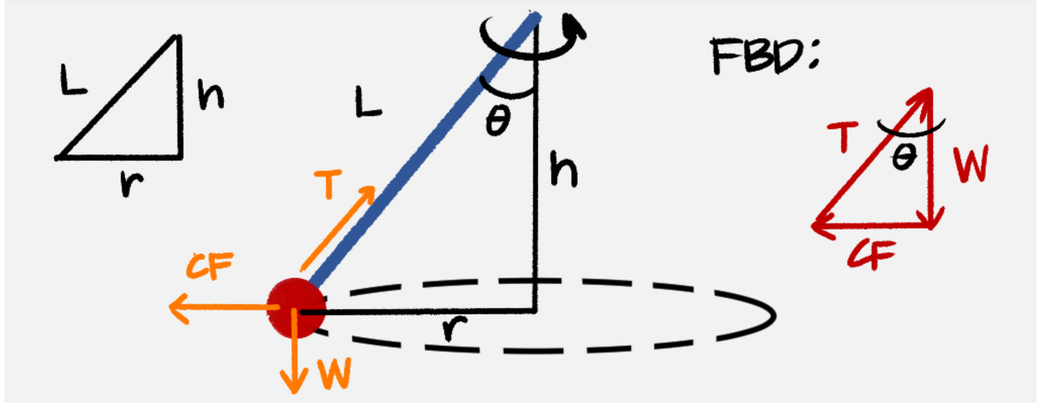
ANAYLSIS

$$F_R + W\sin\theta = 0$$
$$F_R = -mg\sin\theta$$
$$\text{TENSION} = mg\cos\theta$$

PERIOD:
$$t = 2\pi\sqrt{\frac{L}{g}} \;;\; \frac{s}{rev}$$

FREQUENCY:
$$f = T^{-1} = \frac{1}{2\pi}\sqrt{\frac{g}{L}} \;;\; \frac{rev}{s}$$

CONICAL PENDULUM

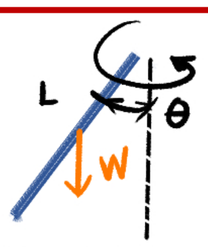


ANAYLSIS

$$\tan\theta = \frac{CF}{W} \qquad \cos\theta = \frac{W}{T}$$
$$CF = m\omega^2 r \qquad \cos\theta = \frac{g}{\omega^2 L}$$
$$W = mg$$
$$\tan\theta = \frac{m\omega^2 r}{mg}$$
$$t = 2\pi\sqrt{\frac{h}{g}}$$
$$\tan\theta = \frac{V^2}{gr}$$

θ DECREASES: h → L

$$t = 2\pi\sqrt{\frac{L}{g}} \rightarrow \text{MAXIMUM TIME FOR REV.}$$

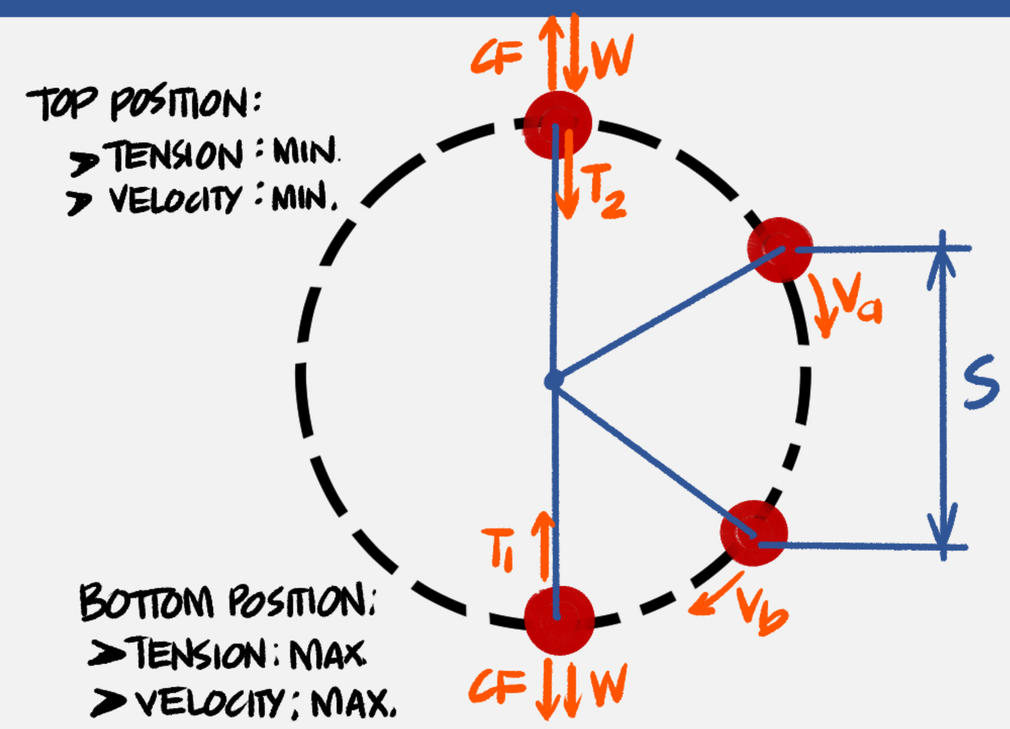


ROTATED ROD OF UNIFORM MASS

$$\cos\theta = \frac{2g}{\omega^2 L}$$

$\omega^2 L > 2g$

VERTICAL CIRCULAR MOTION



NOTE

- AT TOP POSITION, TENSION AND VELOCITY IS MINIMUM
- AT BOTTOM POSITION, TENSION AND VELOCITY IS MAXIMUM

CRITICAL VELOCITY

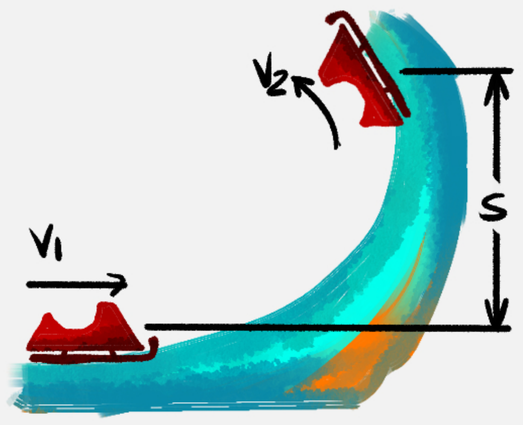
MAXIMUM TOLERABLE VELOCITY AT TOP TO CONTINUE CIRCULAR MOTION

$$V_{CRITICAL} = \sqrt{gR}$$

$$V_b^2 = V_a^2 \pm 2aS$$

V_a = INITIAL VELOCITY
V_b = FINAL VELOCITY
S = VERTICAL DISPLACEMENT
a = (+) ↓ (–) ↑

OBJECTS SLIDING ON A SMOOTH (FRICTIONLESS) SURFACE

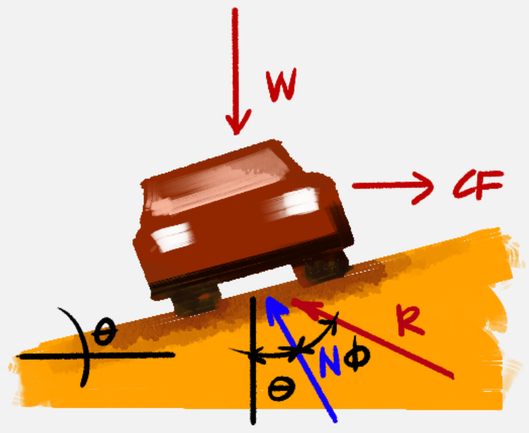


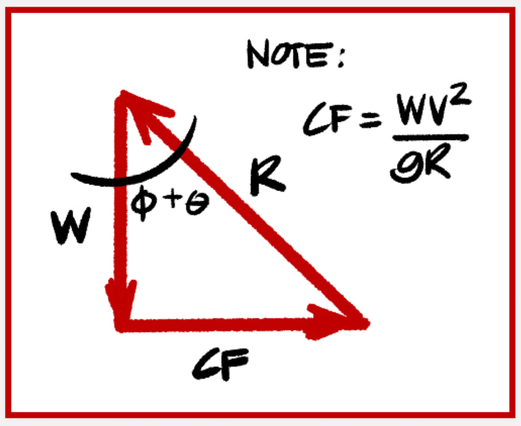
$$V_2^2 = V_1^2 \pm 2aS$$

V_a = INITIAL VELOCITY
V_b = FINAL VELOCITY
S = VERTICAL DISPLACEMENT
a = (+) ↓ (–) ↑

NOTICE THAT FORMULA IS THE SAME AS STRING

BANKING FOR A HI-WAY CURVE





IDEAL ANGLE: FRICTIONLESS ROAD

$$\tan\theta = \frac{\frac{WV^2}{gR}}{W} = \frac{V^2}{gR}$$

$e = \tan\theta$
e = SUPER ELEVATION

FRICTION IS CONSIDERED

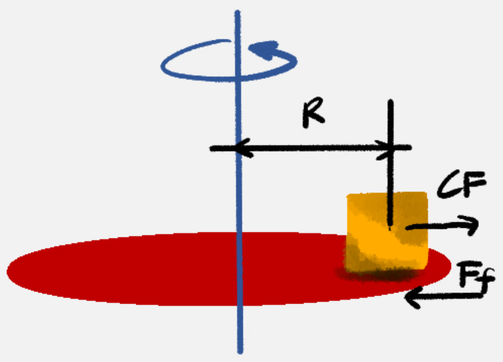
$$\tan(\theta + \phi) = \frac{V^2}{gR}$$

SPECIAL CASE:
➢ CAR IS ON THE POINT OF SLIPPING DOWN

$$\tan(\theta - \phi) = \frac{V^2}{gR}$$

ϕ = ANGLE OF FRICTION

HORIZONTAL ROTATING PLATFORM

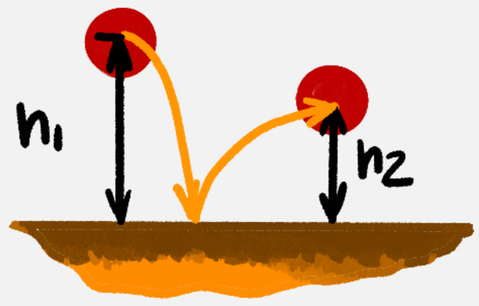


$$\tan\phi = \mu$$

$$\tan\phi = \frac{\omega^2 R}{g}$$

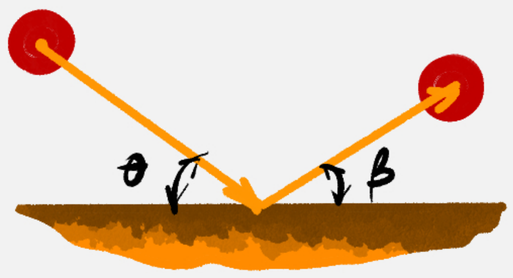
$$\tan\phi = \frac{V^2}{gR}$$

DROPPED AND REBOUNDS



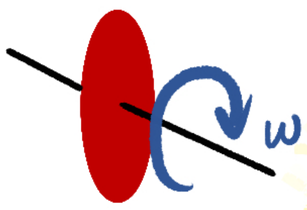
$$e = \sqrt{\frac{h_2}{h_1}}$$

REBOUNDS AT AN ANGLE



$$e = \frac{\tan\beta}{\tan\theta}$$

CENTROIDAL ROTATION



$$I = mk^2 = \frac{W}{g}k^2$$

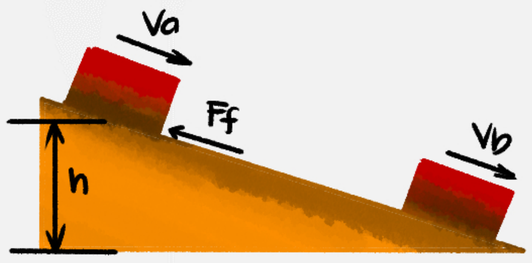
$$KE_R = \frac{1}{2}Iw^2$$

I = MASS MOMENT OF INERTIA

k = POLAR RADIUS OF GYRATION

WORK AND ENERGY

WORK-ENERGY EQ. FOR CONSTANT FORCES



$$KE_A \pm WORK \pm Wh = KE_B$$

$$W = FS \quad | \quad P = \frac{W}{t}$$

$$PE = mgh \qquad KE = \frac{1}{2}mv^2$$

POWER = FORCE × VELOCITY

IMPULSE AND MOMENTUM

IMPULSE IS THE CHANGE IN MOMENTUM

IMPULSE = CHANGE IN MOMENTUM

$$F\Delta t = mV_2 - mV_1$$

IMPULSE

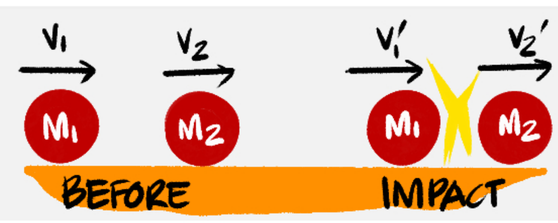
$$I = F\Delta t$$

MOMENTUM

$$P = mV$$

LAW OF CONSERVATION OF MOMENTUM

IN EVERY PROCESS WHERE THE VELOCITY IS CHANGED, THE MOMENTUM LOST BY ONE BODY OR SET OF BODIES IS EQUAL TO THE MOMENTUM GAIN BY ANOTHER BODY OR SET OF BODIES.



$$M_1V_1 + M_2V_2 = M_1V_1' + M_2V_2'$$

COEFFICIENT OF RESTITUTION

$$e = \frac{\text{RELATIVE VEL. AFTER IMPACT}}{\text{RELATIVE VEL. BEFORE IMPACT}} = \frac{V_2' - V_1'}{V_1 - V_2}$$

ELASTIC / INELASTIC COLLISION	$0 < e < 1$
PERFECTLY INELASTIC COLLISION	$e = 0$
PERFECTLY ELASTIC COLLISION	$e = 1$

ENERGY LOSS OF 2 MASSES MOVING IN OPPOSITE DIRECTION

$$\text{LOSS IN KE} = \frac{1}{2}\frac{M_1M_2}{M_1+M_2}(1-e^2)(V_1-V_2)^2$$

ADDITIONAL: MACHINERY'S HANDBOOK FORMULA

POLAR MOMENT OF INTERTIA

$$J_m = \frac{\rho L}{g}J$$

J_m = POLAR MOMENT OF INERTIA

$J = ft^4$; $L = ft$; $\rho = lb/ft^3$

$$J_m = \rho L J$$

$J_m = kg\text{-}m^2$, $\rho = kg/m^3$, $L = m$, $J = m^4$

RADIUS OF GYRATION

$$k_o = \sqrt{\frac{J_m g}{W}} \qquad J_m = \frac{Wk_o^2}{g}$$

$J_m = ft\text{-}lb\text{-}s^2$, $W = lb$, $k_o = ft$

$$k_o = \sqrt{\frac{J_m}{m}} \qquad J_m = mk_o^2$$

$J_m = kg\text{-}m^2$, $\rho = kg/m^3$, $k_o = m$

RADIUS OF OSCILLATION

ISOSCELES TRIANGLE	$\frac{3}{4}h$
CIRCLE	$\frac{5}{8}d$
PARABOLA	$\frac{5}{7}h$

CENTER OF PERCUSSION

Q = DISTANCE FROM AXIS OF ROTATION TO CENTER OF PERCUSSION

$$Q = \frac{k_o^2}{r}$$

FORMULAS FOR TORQUE AND ANGULAR ACCELERATION

$$T_o = J_m\alpha$$

$$T_o = mk_o^2\alpha$$

$$T_o = \frac{Wk_o^2\alpha}{g}$$

T_o = TORQUE ; lb ft

J_m = MOMENT OF INERTIA ; lb-ft-sec²

k_o = RADIUS OF GYRATION ; ft

α = ∠ ACCEL. ; $\frac{rad}{s^2}$

KINETIC ENERGY

$$KE = \frac{1}{2}mv^2$$

$$KE = \frac{1}{2}\cdot\frac{W}{g}v^2$$

TOTAL KINETIC ENERGY: $KE_{TOT} = \frac{1}{2}mv^2 + \frac{1}{2}J_m w^2$

FORCE A BLOW

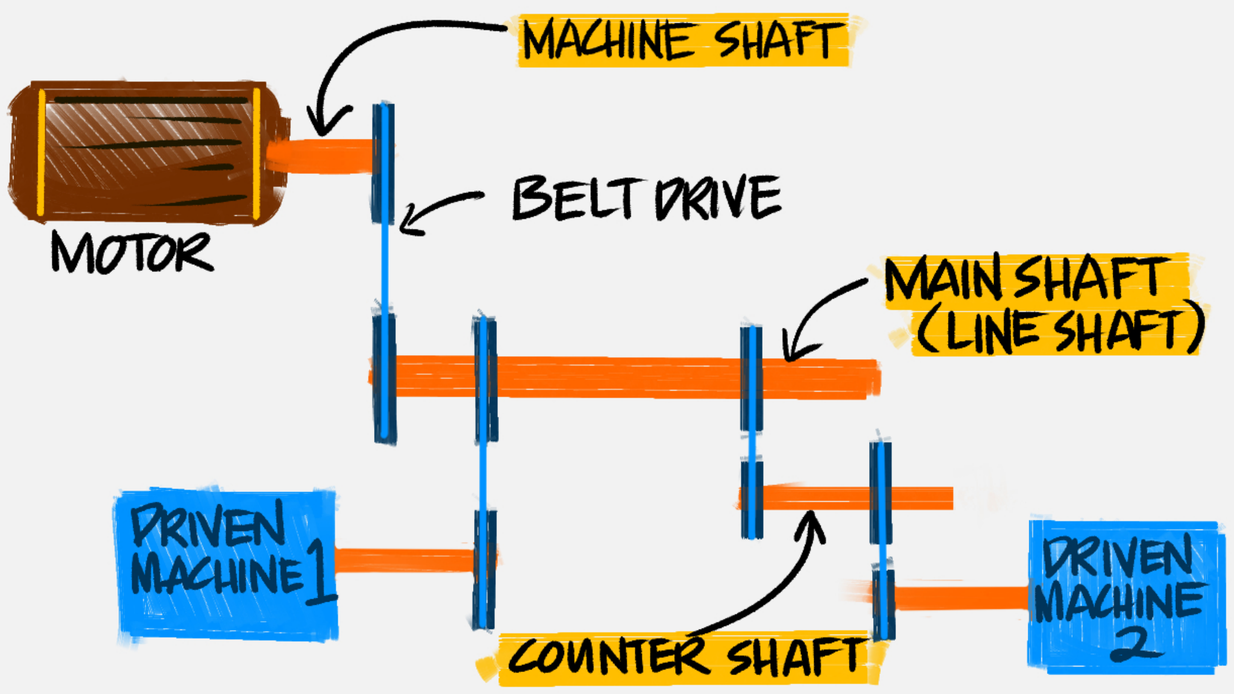
$$\text{AVE. FORCE OF BLOW} = \frac{WS}{d}$$

S = TOTAL HEIGHT, ft,
W = WEIGHT OF DRIVER, lb
d = DISTANCE WHERE PILE, ft. IS DRIVEN

SHAFTS

- A ROTATING MACHINE ELEMENT WHICH IS USED TO TRANSMIT POWER FROM ONE PLACE TO ANOTHER. THE POWER IS DELIVERED BY SOME TANGENTIAL FORCE AND THE RESULTANT TORQUE SETUP WITHIN THE SHAFT PERMITS THE POWER TO BE TRANSFERRED TO VARIOUS MACHINES LINKED UP TO THE SHAFT.

LAYOUT OF POWER TRANSMISSION



STANDARDS

STD. STOCK LENGTHS	16, 20, 24, FT.
MATERIALS FOR TRANSMISSION SHAFTS	COLD – ROLLED, HOT – ROLLED, FORGED CARBON STEEL

STANDARD MACHINERY SHAFTING SIZES	
$\frac{1}{2}'' - 2\frac{1}{2}''$	$\frac{1}{16}''$
$2\frac{5}{8}'' - 4''$	$\frac{1}{8}''$
$4\frac{1}{4}'' - 8''$	$\frac{1}{4}''$

TWO GENERAL CLASSES OF SHAFTS	
MACHINE SHAFT	A SHAFT FORMING AN INTEGRAL PART OF THE MACHINE ITSELF.
TRANSMISSION SHAFT	A SHAFT USED TO TRANSMIT POWER BETWEEN THE SOURCE AND THE MACHINES ABSORBING THE POWER.
TYPES OF TRANSMISSION SHAFTS	
LINE SHAFT (MAIN SHAFT)	IS ONE DRIVEN BY A PRIME MOVER.
COUNTERSHAFTS JACKSHAFTS HEADSHAFTS	ARE SHAFTS BETWEEN A LINE SHAFT AND A DRIVEN MACHINE
AXLE	A STATIONARY MEMBER, PRINCIPALLY LOADED IN BENDING WITH GEARS, PULLEYS AND WHEELS ROTATING ON IT.
SPINDLE	A SHORT SHAFT OR AXLE ON MACHINE

POWER

SI: $P = 2\pi TN$

ENGLISH: $P = \frac{2\pi TN}{33{,}000}$ $P = hp$, $T = lb\,ft$, $N = rpm$

TORQUE: (r of pulley)

$T = Fr$

(BELT) $T = (F_1 - F_2)r$

(GEAR) $T = (F_t)r$ (r = PITCH r OF GEAR)

TORSIONAL STRESS

$S_s = \frac{Tc}{J}$

$S_{st} = \frac{16T}{\pi D^3}$ (SOLID)

$S_{st} = \frac{16T}{\pi D_o^3(1 - B^4)}$ (HOLLOW)

BENDING STRESS

$S_b = \frac{Mc}{I}$

$S_b = \frac{32M}{\pi D^3}$

$S_b = \frac{32M}{\pi D_o^3(1 - B^4)}$

$J = \frac{\pi}{32} D^4$

$I = \frac{\pi}{64} D^4$

TORSIONAL DEFLECTION

$\theta = \frac{TL}{JG}$

θ = DEFLECTION IN RADIANS
T = TORQUE
L = LENGTH OF SHAFT
J = POLAR MOMENT OF INERTIA
G = MODULUS OF RIGIDITY IN SHEAR, 11.5×10^6 psi (STEEL)

FROM MACHINERY'S HANDBOOK

$\theta = \frac{584TL}{D^4 G}$

θ = DEF IN DEGREES
T = lb-in
L = in
D = ø, in
G = 11,500 psi

ALLOWABLE TWISTING MOMENT

FOR ANY CROSS - SECTION

$T = \frac{63{,}000P}{N}$

P = POWER in hp
N = SPEED, rpm

DIAMETER OF SOLID CIRCULAR SHAFT

$D = \sqrt[3]{\frac{5.1T}{S_{st}}} = \sqrt[3]{\frac{321{,}000P}{NS_s}}$

T = lb-in.
S_{st} = psi

ASME CODE

SHEAR DESIGN STRESS	$S_{sd} = 0.3S_{yt}$	$S_{sd} = 0.18S_{ut}$
NORMAL DESIGN STRESS	$S_{sd} = 0.6S_{yt}$	$S_{sd} = 0.36S_{ut}$
SHAFT WITH KEYWAY	$S_d = 0.75(S_{d\,SHEAR/NORMAL})$	

COMBINED TORSION AND BENDING

$$S_s = \frac{16}{\pi D^3}\sqrt{(k_m M)^2 + (k_t T)^2}$$

IF NOT GIVEN, THE ALLOWABLE STRESS FROM ASME CODE (COMMERCIAL SHAFTFS)

FOR SIMPLE TORSION	
SHAFT W/O KEYWAY	8,000 psi
SHAFT W/ KEYWAY	6,000 psi
FOR SIMPLE BENDING	
SHAFT W/O KEYWAY	16,000 psi
SHAFT W/ KEYWAY	12,000 psi

k_t = COMBINED SHOCK & FATIGUE FACTOR FOR TORSION

k_m = COMBINED SHOCK & FATIGUE FACTOR FOR BENDING

COMBINED AXIAL, BENDING, AND TORSION (FOR HOLLOW)

$$S = \frac{16}{\pi D_o^3 (1-B^4)} \sqrt{\left[K_m M + \alpha \frac{F D_o}{8}(1+B^2)^2\right]^2 + (K_t T)^2}$$

$B = \frac{D_i}{D_o}$ $\quad \alpha$ = COLUMN FACTOR, S_y = COMPRESSION YIELD STRESS

$k = \text{RADIUS OF GYRATION} = \sqrt{\frac{I}{A}}$ | $k = D/4$ SOLID; $k = (D_o/4)(1+B^2)$ HOLLOW

VALUE OF n	
HINGED ENDS	1.0
FIXED ENDS	2.26
WITH BEARING SUPPORTS	1.6

COLUMN FACTOR

TENSILE LOAD	$\alpha = 1.0$
FOR COMPRESSION LOAD	
STRAIGHT LINE FORMULA $(L/k) \le 115$	$\alpha = \frac{1}{1 - 0.0044(\frac{L}{K})}$
EULER'S EQN. $(\frac{L}{K}) > 115$	$\alpha = \frac{S_y}{\pi^2 n E}(\frac{L}{K})^2$

PSME CODE

	STRESS	Nsf	FORMULA
MAIN POWER TRANSMITTING SHAFTS	4,000 psi	15	$P = \frac{D^3 N}{80}$ | P = hp, D = INCHES, N = rpm
LINE SHAFTS CARRYING PULLEYS	6,000 psi	10	$P = \frac{D^3 N}{53.5}$ | P = hp, D = INCHES, N = rpm
SMALL SHORT SHAFTS	8,500 psi	7	$P = \frac{D^3 N}{38}$ | P = hp, D = INCHES, N = rpm

MAXIMUM VERTICAL SHEAR STRESS OF SHAFTS

- CIRCULAR CROSS-SECTION: $S_v = 4V/3A = 16V/3\pi D^2$
- RECTANGULAR CROSS-SECTION: $S_v = \frac{3V}{2A}$ | V = MAXIMUM SHEAR FORCE

SHORTCUT: HOLLOW SHAFT W/ SAME MAT'L.

GENERAL FORMULA: $ax^2 + bx + c = 0$

$a = 2$, $b = -D/x$, $c = -xD^2$ | D = DIAMETER; X = PERCENT REDUCTION IN WEIGHT (50% = 0.5)

50%: $D_o = \left(\frac{1+\sqrt{2}}{2}\right)D$ | D_o = OUTSIDE Ø OF HOLLOW

DEFLECTIONS (MACHINERY'S HANDBOOK)

ANGLE OF TWIST "NOT EXCEEDING 0.08° PER FOOT LENGTH OF SHAFT"	ENGLISH: $D = 0.29\sqrt[4]{T}$ D = Ø; in. T = lb-in	$D = 4.6\sqrt[4]{\frac{P}{N}}$ P = hp N = rpm
	SI UNITS: $D = 2.26\sqrt[4]{T}$ D = Ø; mm T = N-mm	$D = 125.7\sqrt[4]{\frac{P}{N}}$ P = WATTS N = rpm
SHAFT DEFLECTION 1 DEGREE FOR $L = 20 \times D$	ENGLISH: $D = 0.10\sqrt{T}$ D = Ø; in. T = lb-in	$D = 4.0\sqrt{\frac{P}{N}}$ P = hp N = rpm

SHAFT LINEAR DEFLECTION

NO BENDING ACTION EXCEPT ITS WEIGHT	$L = 8.95\sqrt[3]{D^2}$
BENDING ATION OF PULLEYS	$L = 5.2\sqrt[3]{D^2}$

L = LENGTH in ft
D = Ø; in.

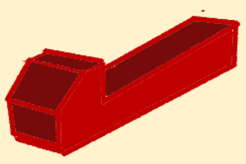
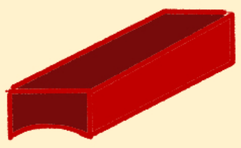
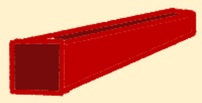
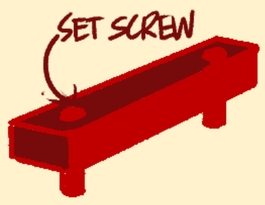
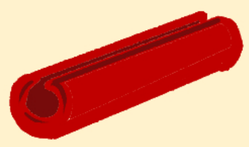
KEYS

KEY

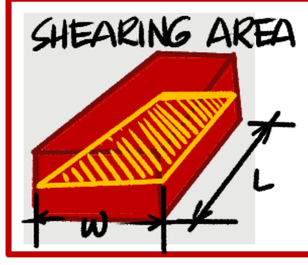
KEYWAY

A **MILD STEEL** INSERTED BETWEEN THE SHAFT AND HUB TO **CONNECT THESE TOGETHER IN ORDER TO PREVENT RELATIVE MOTION BETWEEN THEM.** INSERTED PARALLEL TO THE AXIS OF THE SHAFT.

IS A **SLOT OR RECESS IN A SHAFT AND HUB OF THE PULLEY** TO ACCOMMODATE A KEY.

CONSTANT CROSS-SECTIONAL	**FLAT KEY**	IS **RECTANGULAR** IN SECTION	
CONSTANT CROSS-SECTIONAL	**SQUARE KEY**	**SQUARE** IN SECTION HALF OF ITS DEPTH SUNK IN THE SHAFT	
CONSTANT CROSS-SECTIONAL	**ROUND KEY**	**CIRCULAR** CROSS SECTION	
CONSTANT CROSS-SECTIONAL	**BARTH KEY**	SQUARE KEY WITH BOTTOM TWO CORNERS BEVELLED.	
VARIABLE CROSS-SECTIONAL	**WOODRUFF KEY**	HALF OF CIRCULAR DISK - RECTANGULAR (FEMALE) - SEMI-CYLINDRICAL (MALE)	
VARIABLE CROSS-SECTIONAL	**GIB-HEAD TAPER KEY**	FLAT KEY W/ **SPECIALIZED GIB HEAD** FOR EASY DRIVING AND REMOVAL	
VARIABLE CROSS-SECTIONAL	**SADDLE KEY**	USED **WITHOUT KEYWAY** ON THE SHAFT.	
OTHER KEYS	**KENNEDY KEY (TANGENTIAL KEY)**	TAPERED SQUARE KEY, W/ OR W/O GIB HEADS, ASSEMBLED WITH THE DIAGONAL DIMENSION IN A CIRCUMFERENTIAL DIRECTION.	
OTHER KEYS	**FEATHER KEY**	ALLOWS THE HUB TO MOVE BUT PREVENTS ROTATION ON THE SHAFT. - ONE W/C HAS A TIGHT FIT INTO ONE MEMBER, LOOSE SLIDING FIT IN THE OTHER	
OTHER KEYS	**ROLL PIN**	DRIVEN OR PRESSED INTO A HOLE THAT IS **SMALL ENOUGH TO CLOSE** THE SLIT, ASSEMBLED IN RADIAL DIRECTION.	

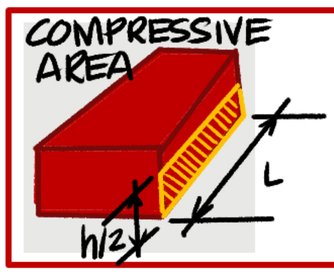
SHEARING STRESS



$$T = F \cdot r = F \times \left(\frac{D}{2}\right)$$

$$S_s = \frac{F}{A} = \frac{2T}{WLD}$$

COMPRESSIVE



$$T = F \cdot r = F \times \left(\frac{D}{2}\right)$$

$$S_s = \frac{F}{A} = \frac{4T}{hLD}$$

IDEAL KEY

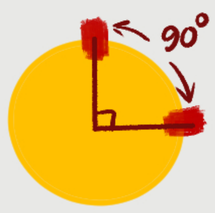
- EQUALLY STRONG IN BOTH CRUSHING AND SHEARING, THE TORQUE IS EQUAL

$$T = \frac{S_s D w L}{2} = \frac{S_c D h L}{4} \quad \Big| \quad \frac{2S_s}{S_c} = \frac{h}{w}$$

KEYS MADE OF SAME MAT'L.

$$W = \frac{D}{4} \quad | \quad L = 1.2D$$

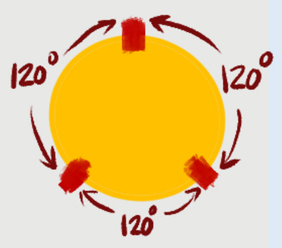
DOUBLE KEYED SHAFT



T = TORQUE CAPACITY OF ONE KEY
T_t = TOTAL TORQUE CAPACITY

$$T = \frac{T_t}{2} \times 1.2$$

TRIPLE KEYED SHAFT



T = TORQUE CAPACITY OF ONE KEY
T_t = TOTAL TORQUE CAPACITY

$$T = \frac{T_t}{3} \times 1.15$$

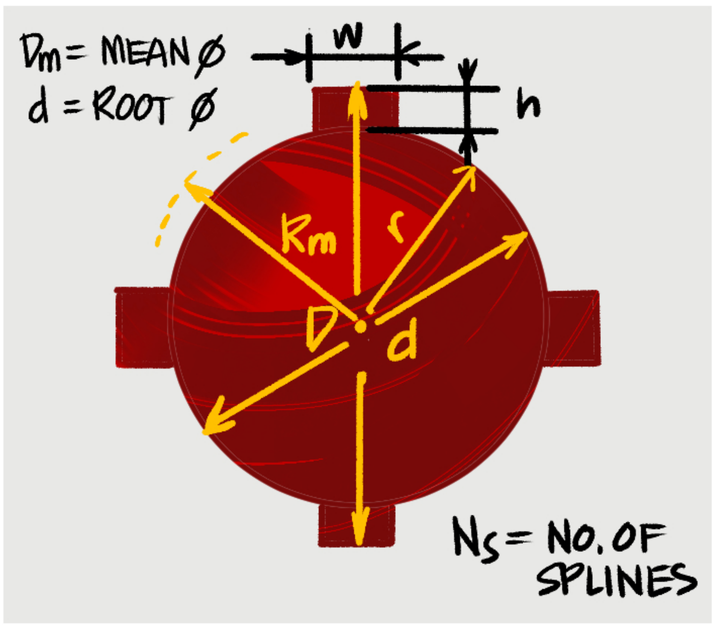
SMOOTH LOADING	1.5
MINOR SHOCK	2 - 2.25
SEVER SHOCK	UP TO 4.5

SPLINES

- IS A KEY MADE INTEGRAL WITH THE SHAFT WHICH FITS IN THE KEYWAYS BROACHED IN THE HUB.
- USUALLY CONSISTS OF **4, 6, 8, 10**

$$P = 2\pi TN$$

$$R_m = \frac{1}{2}\left(\frac{D+d}{2}\right)$$



SHEARING OF SPLINES

$$S_s = \frac{F}{A} = \frac{F}{WLN_s} = \frac{2T}{(WL)dN_s}$$

COMPRESSION OF SPLINES

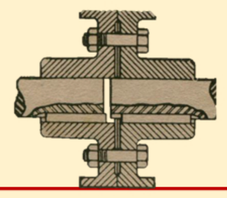
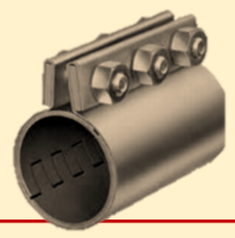
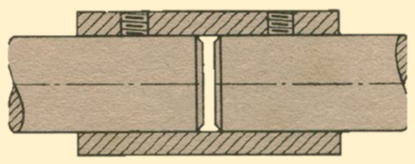
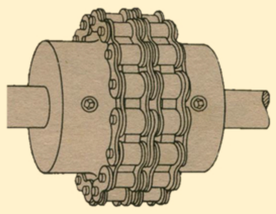
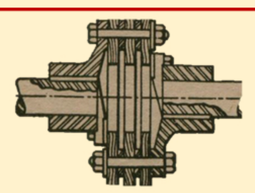
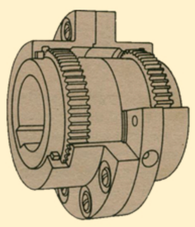
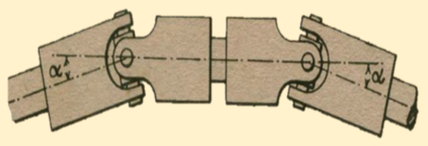
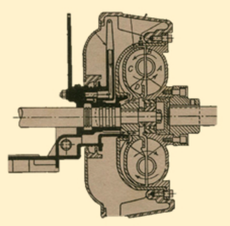
$$S_c = \frac{F}{A} = \frac{F}{hLN_s} = \frac{2T}{(hL)D_mN_s}$$

TORQUE CAPACITY OF ONE SPLINE

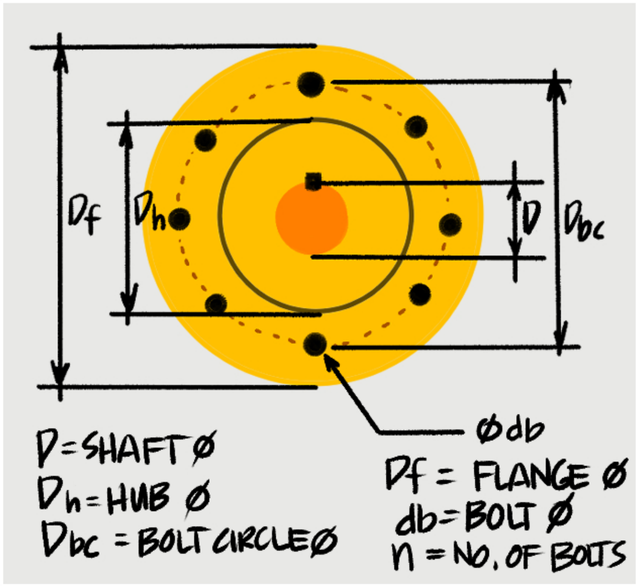
$$T_o = \frac{T}{N_s} \times 1.1$$

COUPLINGS

- IS A MECHANICAL DEVICE WHICH IS USED TO CONNECT LENGTH OF SHAFTS PERMANENTLY

RIGID COUPLINGS	**RIGID COUPLING**	CONNECT TWO SHAFTS WHICH ARE **PERFECTLY ALIGNED.**	
	FLANGE COUPLING	TWO HALVES OF **FLANGES** CONNECTED BY BY BOLTS	
	COMPRESSION COUPLING (CLAMP)	**USES TWO SPLIT CONES** CONNECTED BY BOLTS TO PRODUCE A **WEDGING ACTION** THAT TIGHTENS THE PARTS OF THE COUPLING AND THE SHAFT.	
	COLLAR COUPLING	CONSISTS OF A **CYLINDRICAL COLLAR** PRESSED OVER THE ENDS OF THE TWO COLLINEAR SHAFTS BEING CONNECTED.	
FLEXIBLE COUPLINGS	**CHAIN COUPLING**	CONSISTS ESSENTIALLY OF **TWO CHAIN SPROCKETS CONNECTED** WITH SHORT CONTINUOUS LENGTH OF **ROLLER OR SILENT CHAIN.**	
	OLDHAM COUPLING	JOIN TWO SHAFTS WHICH HAVE **LATERAL MIS-ALIGNMENT.**	
	FLEXIBLE DISK COUPLING	AN ALL METAL COUPLING WITH THE INTERMEDIATE FLEXIBLE ELEMENTS BEING **THIN STEEL DISKS.**	
	GEAR TYPE COUPLING	TWO HUBS WITH **EXTERNAL GEAR TEETH,** CONNECTED BY **FLANGE W/ INTERNAL GEAR TEETH,** FLANGES BOLTED	
	UNIVERSAL JOINT (HOOKE'S COUPLING)	CONNECT SHAFTS WHOSE **AXES INTERSECT** THAT IS, WHOSE **ANGULAR MISALIGNMENT IS PERMANENT.**	
	HYDRAULIC COUPLING	EMPLOYS A **FLUID TO PROVIDE ANGULAR FLEXIBILITY** BETWEEN THE INPUT AND OUTPUT SHAFTS.	

FLANGE COUPLINGS



POWER TRANSMITTED

$$P = 2\pi TN \quad ; kW$$
$$P = \frac{2\pi TN}{33000} \quad ; hp$$

TOTAL FORCE TRANSMITTED

$$F = T/(Dbc/2) \quad | \quad \text{FORCE PER BOLT } Fb = F/n$$

SHEARING OF BOLTS

$$S = \frac{F}{A} = \frac{2T}{\frac{\pi}{4}db^2 \cdot Dbc \cdot n} = \frac{8T}{\pi db^2 Dbc n}$$

(Ødb)

COMPRESSIVE STRESS BET. BOLT AND FLANGE

$$S = \frac{F}{A} = \frac{2T}{(db\,t_f)Dbc \times n}$$

NOTE:

SAFE TORQUE IS LOWEST!

TAPER PINS

STRENGTH OF TAPER PINS: ENGLISH

$$d = 1.13\sqrt{\frac{T}{DS}} \qquad d = 283\sqrt{\frac{HP}{NDS}}$$

$$S = \frac{1.27T}{Dd^2}$$

D= SHAFT Ø
d = TAPERED PIN Ø

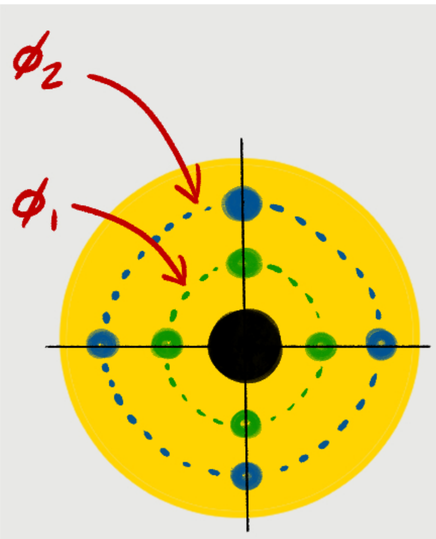
STRENGTH OF TAPER PINS: SI UNITS

$$d = 110.3\sqrt{\frac{P}{NDS}}$$

d, D } mm
P = WATTS
S = N/mm²
N = rpm

S = SAFE UNIT STRESS, psi
HP = HORSEPOWER
N = SPEED, rpm

MULTIPLE CONCENTRIC BOLT CIRCLES



TORQUE

$$T = F_1R_1n_1 + F_2R_2n_2 + \ldots$$

RELATIONSHIPS

$$\frac{\gamma_1}{R_1} = \frac{\gamma_2}{R_1} \quad \text{(SHEAR STRAIN)}$$

$$\frac{S_{s1}}{GR_1} = \frac{S_{s2}}{G_2R_2} \quad \text{(SHEAR STRESS)}$$

$$\frac{F_1}{R_1} = \frac{F_2}{R_2}$$

PRESSURE VESSELS

MATERIALS FOR PRESSURE VESSLES

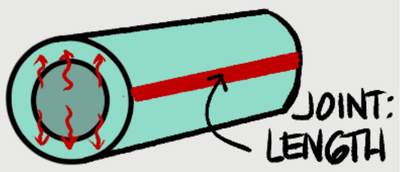
CARBON AND LOW-ALLOY STEELS | NON-FERROUS ALLOYS | HIGH-ALLOY STEELS | CAST IRON | FERRITIC STEELS

THIN-WALLED PRESSURE VESSEL

VALLANCE: $\frac{t}{di} \leq 0.07$ | COLLINS: $t \leq 0.1 di$ | $Pi < \frac{1}{6} S_T$

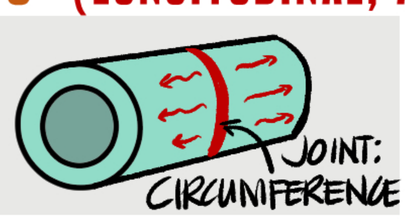
CYLINDERS

TTCH (TENSILE, TANGENTIAL, CIRCUMFERENTIAL, HOOP)



$$S_t = \frac{P_i d_i}{2te}$$

LAG (LONGITUDINAL, AXIAL, GIRTH)



$$S_a = \frac{P_i d_i}{4te}$$

SPHERES

$$S = \frac{P_i d_i}{4te}$$

JOINT EFFICIENCY

$$e = \frac{\text{MINIMUM STRENGTH OF JOINT}}{\text{STRENGTH OF SOLID PLATE}}$$

FACTOR OF SAFETY: SPECIFIED MAT'L STRENGTH = 5

THICK-WALLED CYLINDER

VALLANCE: $\frac{t}{di} > 0.07$ | COLLINS: $t > 0.1 di$ | $P_i > \frac{1}{6} S_T$

LAME'S FORMULA (GENERAL FORMULA)

AXIAL STRESS

$$S_a = \frac{P_i r_i^2 - P_o r_o^2}{r_o^2 - r_i^2}$$

TANGENTIAL STRESS

$$S_t = \frac{P_i r_i^2 - P_o r_o^2}{r_o^2 - r_i^2} + \frac{r_i^2 r_o^2}{r^2}\left(\frac{P_i - P_o}{r_o^2 - r_i^2}\right)$$

RADIAL STRESS ← COMPRESSIVE IN RADIAL DIRECT.

$$S_t = \frac{P_i r_i^2 - P_o r_o^2}{r_o^2 - r_i^2} - \frac{r_i^2 r_o^2}{r^2}\left(\frac{P_i - P_o}{r_o^2 - r_i^2}\right)$$

NOTE: "r" → RADIUS OF OUTSIDE OR INSIDE

LAME'S EQUATION	MATERIAL	EQUATION
BRITTLE MATERIALS	CAST IRON	$t = \frac{D_i}{2}\left[\sqrt{\frac{S_t + P_i}{S_t - P_i}} - 1\right]$
DUCTILE MATERIALS	CARBON STEEL, NICKEL ALLOYS	$t = \frac{D_i}{2}\left[\sqrt{\frac{S_t}{S_t - 2P_i}} - 1\right]$

MAXIMUM SHEAR STRESS THEORY

$$t = \frac{D_i}{2}\left[\sqrt{\frac{S_{st}}{S_{st} - P_i}} - 1\right]$$

S_{st} = SHEAR STRESS
S_t = TANGENTIAL
$S_{st} = \frac{S_t}{2}$

MAXIMUM-STRAIN THEORY

BIRNIE'S EQUATION

FOR OPEN-END CYLINDERS

$$t = \frac{D_i}{2}\left[\sqrt{\frac{S_t + (1-\mu)P_i}{S_t - (1+\mu)P_i}} - 1\right]$$

CLAVARINO'S EQUATION

FOR CLOSE-END CYLINDERS

$$t = \frac{D_i}{2}\left[\sqrt{\frac{S_t + (1-2\mu)P_i}{S_t - (1+\mu)P_i}} - 1\right]$$

MAX. ENERGY OF DISTORTION THEORY

OCTAHEDRAL SHEAR STRESS THEORY

$$t = \frac{D_i}{2}\left[\sqrt{\frac{S_t}{S_t - \sqrt{3}P_i}} - 1\right]$$

BARLOWE'S EQUATION

FOR HIGH-PRESSURE PIPE

$$S_t = \frac{P_i d_o}{2t}$$

TANGENTIAL STRESS (INSIDE)	$r = r_i$
$S_{ti} = \frac{P_i(r_o^2 + r_i^2) - 2P_o r_o^2}{r_o^2 - r_i^2}$	
TANGENTIAL STRESS (OUTSIDE)	$r = r_o$
$S_{to} = \frac{2P_i r_i^2 - P_o(r_o^2 + r_i^2)}{r_o^2 - r_i^2}$	
TANG. AND RADIAL WITH ZERO EXTERNAL PRESSURE	$P_o = 0$
MAX. TANG. AND RADIAL WITH ZERO EXTERNAL PRESSURE $S_t, S_r \propto \frac{1}{r}$	$P_o = 0$, $r = r_i$
$S_{t_{MAX}} = \frac{P_i(r_o^2 + r_i^2)}{(r_o^2 - r_i^2)}$	$S_{r_{MAX}} = -P_i$

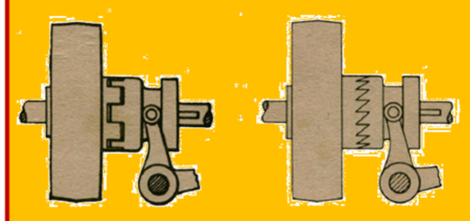
MAXIMUM SHEAR STRESS AT INSIDE

$$S_{S_{t_{MAX}}} = (P_i - P_o)\left[\frac{r_o^2}{r_o^2 - r_i^2}\right]$$

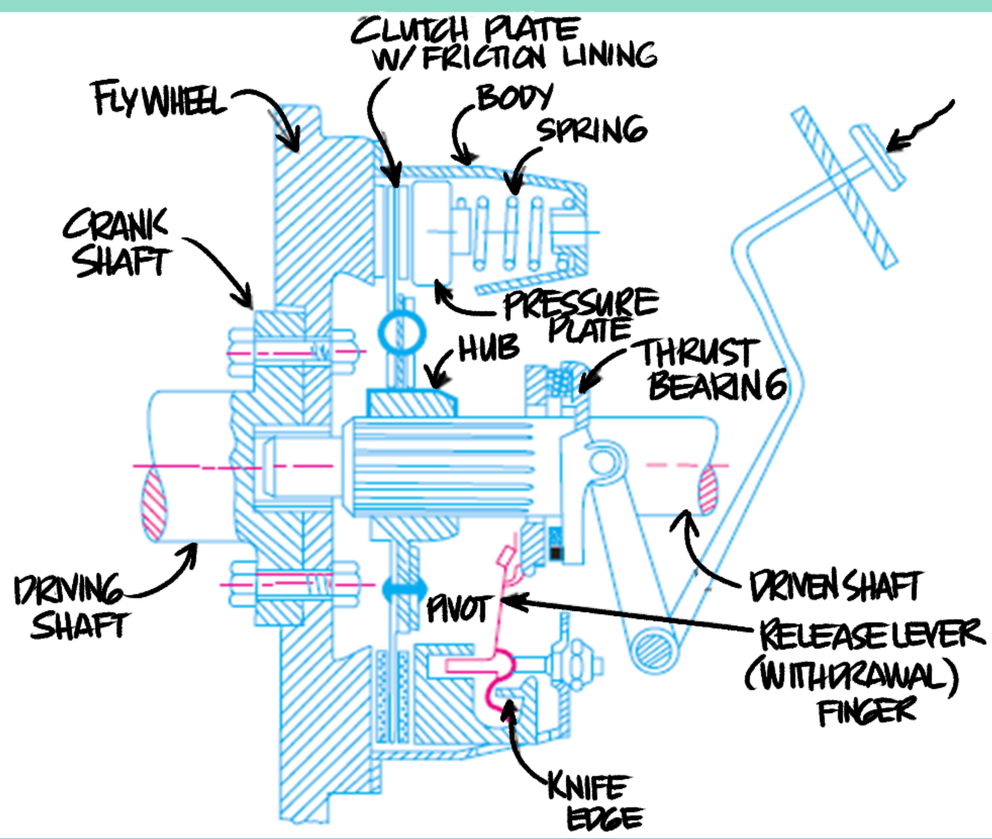
CLUTCHES

USED TO CONNECT SHAFTS SO THAT THE DRIVEN SHAFT ROTATE WITH THE DRIVING SHAFT AND TO **DISCONNECT THEM AT WILL.**

TYPES OF CLUTCHES

JAW CLUTCHES 	JAWS OR TEETH IN THE TWO ELEMENTS INTERLOCK. **TYPES:** • SQUARE/JAW CLUTCH • SPIRAL/BEVEL JAW
FRICTION CLUTCHES	DEPEND UPON FRICTIONAL CONTACT BETWEEN TWO OR MORE MEMBERS FOR TORQUE TRANSMISSION. **TYPES:** • PLATE/DISK CLUTCH • CONE CLUTCH • BAND CLUTCH • BLOCK CLUTCH • EXPANDING RING-CLUTCH
HYDRAULIC CLUTCHES	THE TORQUE IS TRANSMITTED BY A MOVING FLUID
ELECTROMAGNETIC CLUTCHES	THE TORQUE IS TRANSMITTED BY MEANS OF A MAGNETIC FIELD

DISK CLUTCH



FOR MULTIPLE DISK CLUTCHES;
NO. OF PAIRS → $n = N_T - 1$ | N_T = TOTAL NUMBER OF DISK CLUTCHES

	UNIFORM PRESSURE (NEW)	UNIFORM WEAR (OLD)
MEAN FRICTION RADIUS	$r_m = \frac{2}{3}\left(\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2}\right)$	$r_m = \frac{1}{2}(r_o + r_i)$
FORCE (AXIAL)	$F_a = P \times \pi(r_o^2 - r_i^2)$	$F_a = C \times 2\pi(r_o - r_i)$
TORQUE	$T = \frac{2\pi}{3} \times f \times P_N (r_o^3 - r_i^3)$	$T = \pi f \times C \times (r_o^2 - r_i^2)$

POWER

$$P = 2\pi TN$$

TORQUE

$$T = [f F_a r_m] \times n$$

F_a = AXIAL FORCE
r_m = MEAN FRICTION RADIUS

VALUE OF C:

$C = P_{MAX} r_i$ → USE THIS!!
$C = P_{MIN} r_o$
$C = P_m r_m$

CONE CLUTCH

$$r = \frac{r_o + r_i}{2}$$

$$a = r_o - r_i$$

$$\sin\theta = \frac{a}{b}$$

$$\sin\theta = \frac{r_o - r_i}{b}$$

θ = CLUTCH ANGLE ≈ 7.5° – 30°

COEFFICIENT OF FRICTION

$$f = \frac{F_t}{F_N}$$

F_t = FORCE TRANSMITTED

NORMAL FORCE

$$\sin\theta = \frac{F_N}{F_A}$$

TORQUE

$$T = F_t \times r_f = f\left(\frac{F_A}{\sin\theta}\right) \times r_f$$

BRAKES

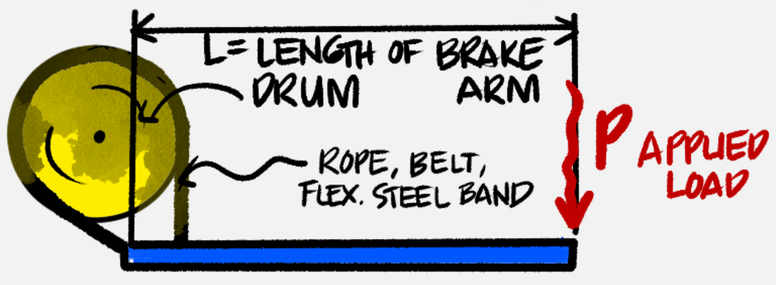
ARE FRICTION DEVICES USED TO **REGULATE OR STOP THE MOTION OF BODIES** (SLOWING THEM DOWN, HOLDING THEIR SPEED CONSTANT, HOLDING THEM AT REST, ETC.

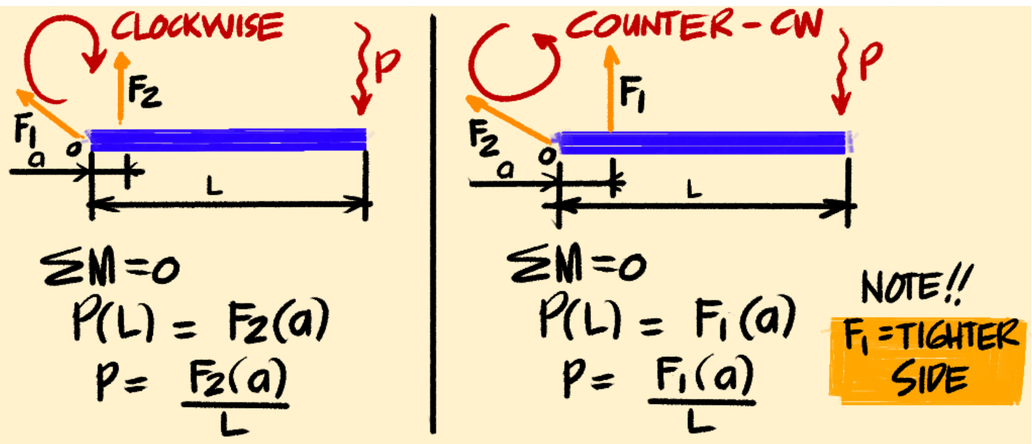
TYPES OF MECHANICAL BRAKES

RADIAL BRAKES (BAND OR BLOCK BRAKES)	• THE FORCE ACTING IS IN **RADIAL** DIRECTION.
AXIAL BRAKES (DISC AND CONE BRAKES)	• THE FORCE ACTING IS IN **AXIAL** DIRECTION. , SIMILAR TO CLUTCHES.

- **SINGLE BLOCK/SHOE BRAKE**
  - ONE BLOCK SHOE IS PRESSED AGAINST THE RIM OF A WHEEL
- **DOUBLE BLOCK/SHOE BRAKE**
  - TWO BLOCKS APPLIED AT THE OPPOSITE ENDS OF THE WHEEL WHICH ELIMINATES THE UNBALANCED FORCE ON THE SHAFT.
- **BAND BRAKE**
  - A FLEXIBLE BAND OF LEATHER WHICH EMBRACES A PART OF THE CIRCUMFERENCE OF THE DRUM.
- **SIMPLE BAND BRAKE**
  - END OF THE BAND IS **ATTACHED TO A FIXED PIN** OF THE LEVER WHILE THE OTHER END IS ATTACHED TO THE LEVER **DIFFERENTIAL BAND BRAKE**
  - IN WHICH THE **TENSION IN THE BAND ASSISTS** IN APPLYING THE BRAKE.

BAND BRAKES





TENSION IN THE BAND

$$\frac{F_1}{F_2} = e^{f\theta}$$

F_1 = FORCE ON TIGHT SIDE
f = COEFF. OF FRICTION
θ = ANGLE OF CONTACT, rad

BRAKE TORQUE

$$T = (F_1 - F_2)r$$

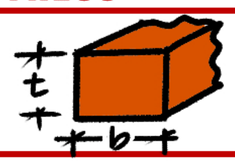
r = RADIUS OF FRICTION SURFACE ON DRUM

MAX PRESSURE

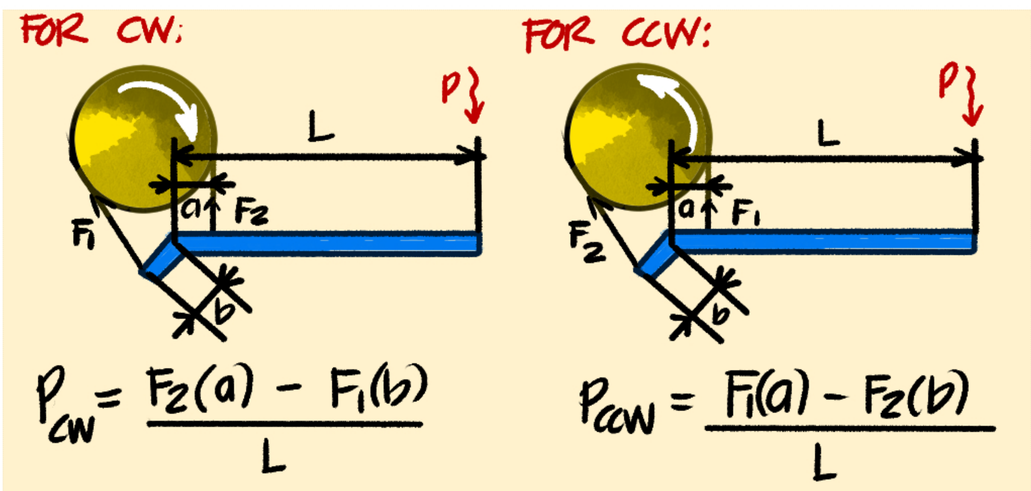
$$P_{max} = \frac{F_1}{br}$$

BAND MAX. STRESS

$$S_b = \frac{F_1}{bt}$$



DIFFERENTIAL BAND BRAKES



HEAT DISSIPATED IN BRAKES (VALLANCE)

$$H = fF_rV$$

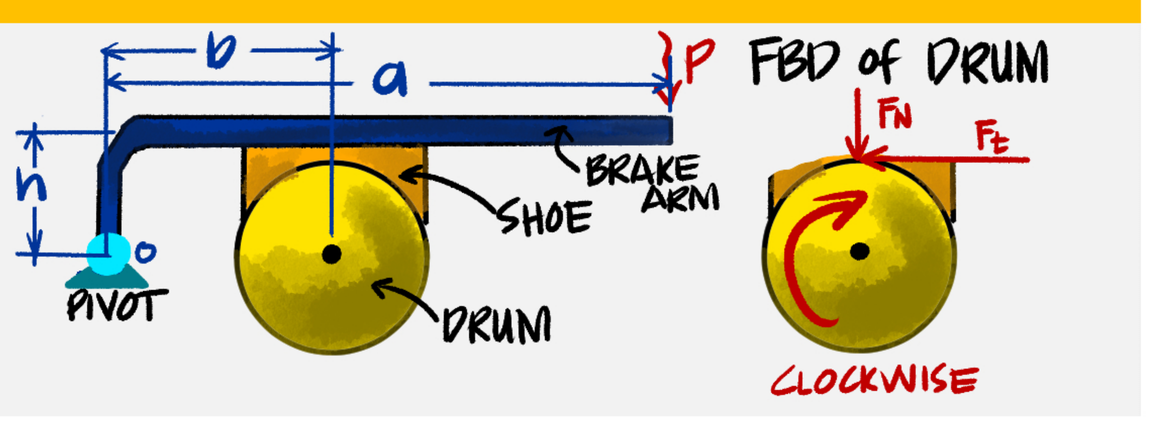
H = HEAT DISSIPATED
F_r = RADIAL FORCE
V = SURFACE VELOCITY

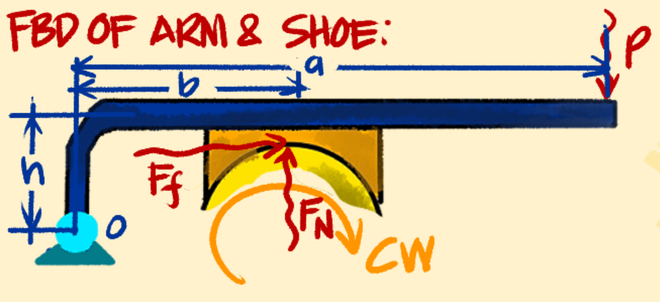
FOR LOWERING BRAKE

$$H = Wh$$

W = WEIGHT ; h = TOTAL DISTANCE

BLOCK BRAKES



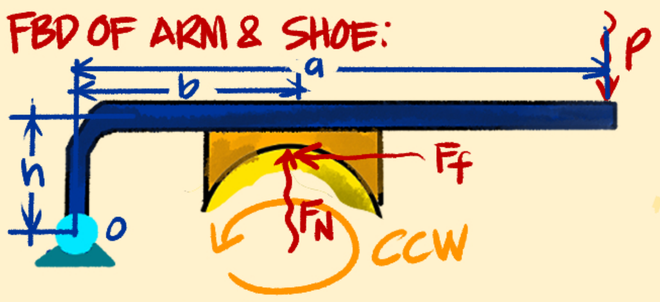


$F_f = fF_N$

$P(a) + (fF_N)(h) = F_N(b)$

$$P = \frac{F_N(b - fh)}{a}$$

IF $fh > b$ → SELF-LOCKING SELF-ENERGIZING

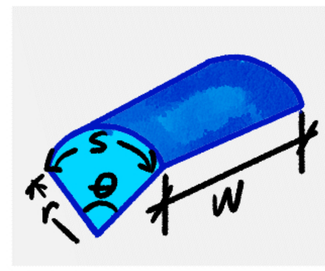


$F_f = fF_N$

$P(a) = F_N(b) + fF_N(h)$

$$P = \frac{F_N(b + fh)}{a}$$

IF $\theta \leq 60°$
→ UNIFORM NORMAL PRESSURE



$F_N = P_N \times A$

$A = S \times W$

WHERE:

$S = r\theta$

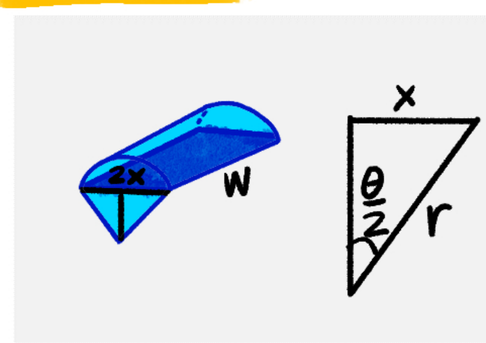
$F_N = P_N \times (r\theta)W$

$T = F_f \times r$

$F_f = fF_N$

$T = f \times P_N \times (r\theta W) \times r$

IF $\theta > 60°$
UNUNIFORM PRESSURE



$F_N = P_{MAX}(A)$

$A = 2xW$

WHERE;

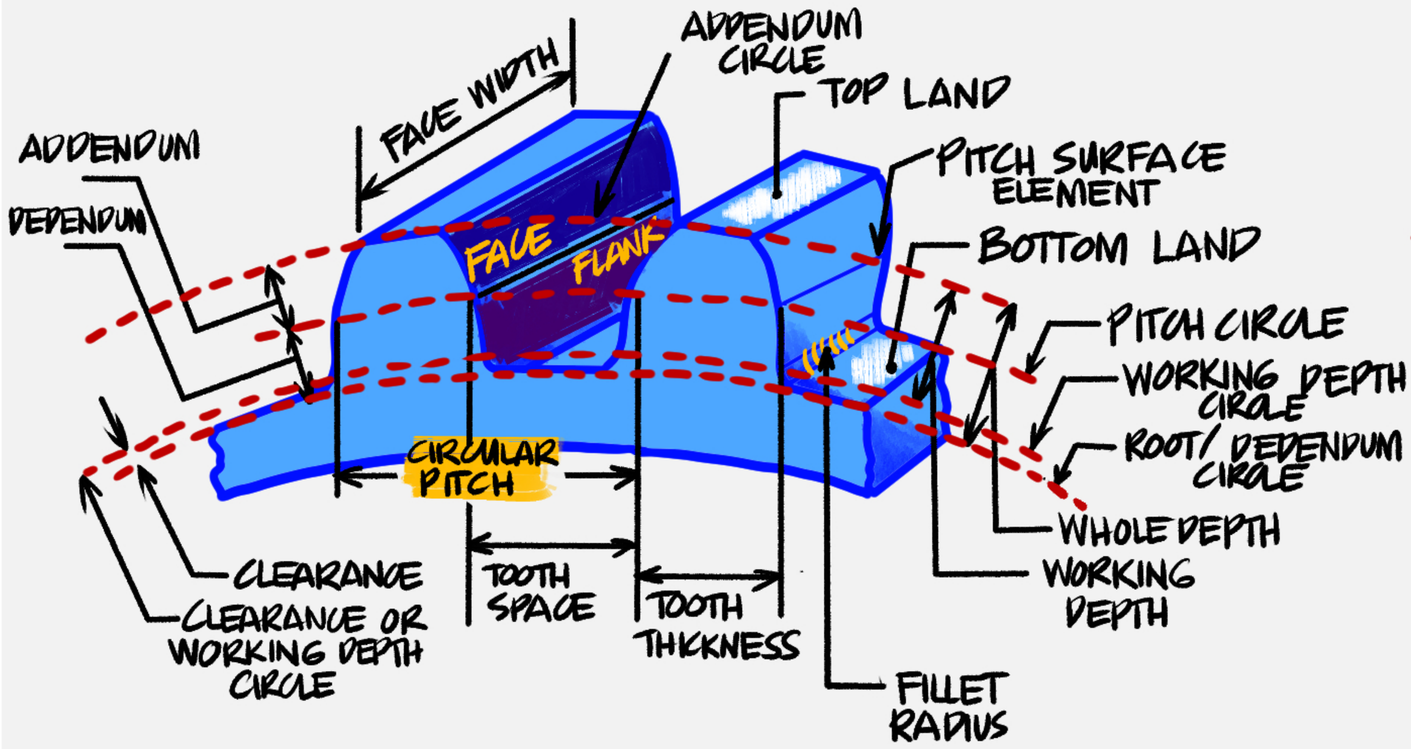
$x = r\sin\left(\frac{\theta}{2}\right)$

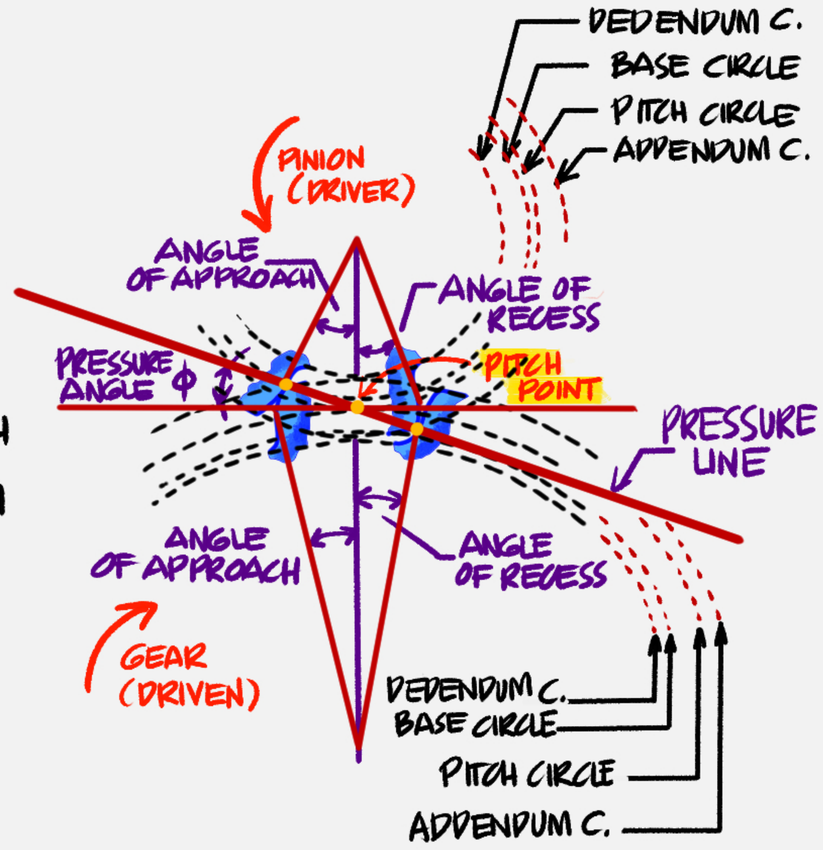
$F_N = P_{MAX}\left(2r\sin\left(\frac{\theta}{2}\right) \times W\right)$

SPUR GEARS

- TOOTHED WHEELS ARE **STRAIGHT AND PARALLEL TO THE SHAFT AXIS**; THEY ARE USED TO TRANSMIT MOTION AND POWER **BETWEEN PARALLEL SHAFTS.**

SPUR GEAR: NOMENCLATURE





PITCH CIRCLE	IMAGINARY CIRCLE, BY PURE ROLLING ACTION, SAME MOTION AS THE ACTUAL GEAR
PITCH CIRCLE DIAMETER	DIAMETER OF THE PITCH CIRCLE. GEAR SIZE IS BY THE PITCH CIRCLE DIAMETER.
PITCH POINT	POINT OF TANGENCY OF THE PITCH CIRCLES
PITCH SURFACE	SURFACE OF THE ROLLING DISCS
ADDENDUM	RADIAL DISTANCE BETWEEN THE PITCH CIRCLE AND THE ADDENDUM CIRCLE
DEDENDUM	RADIAL DISTANCE FROM THE PITCH CIRCLE TO THE ROOT CIRCLE (BOTTOM OF TOOTH SPACE)
ADDENDUM CIRCLE (OUTSIDE CIRCLE)	CIRCLE THAT BOUNDS THE OUTER ENDS OF THE TEETH.
DEDENDUM CIRCLE (ROOT CIRCLE)	CIRCLE THAT BOUNDS THE BOTTOMS OF THE TEETH.
WHOLE DEPTH (TOTAL DEPTH)	RADIAL DISTANCE BETWEEN THE ADDENDUM AND THE DEDENDUM. THE SUM OF THE ADDENDUM AND DEDENDUM
WORKING DEPTH	RADIAL DISTANCE FROM THE ADDENDUM CIRCLE TO THE CLEARANCE CIRCLE. IT IS EQUAL TO THE SUM OF BOTH THE ADDENDUMS OF THE GEARS
CLEARANCE	RADIAL DISTANCE BETWEEN THE WORKING-DEPTH CIRCLE AND THE ROOT CIRCLE. DEDENDUM MINUS THE MATING ADDENDUM
TOOTH THICKNESS	WIDTH ALONG THE PITCH CIRCLE
TOOTH SPACE	WIDTH BETWEEN THE TWO ADJACENT TEETH

BACKLASH	DIFFERENCE BETWEEN THE TOOTH SPACE AND THE TOOTH THICKNESS
FACE OF THE TOOTH	SURFACE BETWEEN THE PITCH CYLINDER AND THE ADDENDUM CYLINDER
FLANK	SURFACE BETWEEN THE PITCH AND ROOT CYLINDERS
TOP LAND	SURFACE OF THE TOP OF THE TOOTH
BOTTOM LAND	SURFACE OF THE BOTTOM OF THE TOOTH SPACE
FACE WIDTH	**LENGTH** OF TEETH IN AN AXIAL DIRECTION
PROFILE	CURVE FORMED BY THE FACE AND FLANK
FILLET RADIUS	CONNECTS THE ROOT CIRCLE TO PROFILE OF TOOTH
PINION	THE SMALLER OF TWO MESHING GEARS
GEAR	THE LARGER OF TWO MESHING GEARS
PATH OF CONTACT	PATH TRACED BY THE POINT OF CONTACT FROM THE **BEGINNING TO THE END OF ENGAGEMENT**
LENGTH OF THE PATH OF CONTACT	LENGTH OF THE COMMON NORMAL CUT-OFF BY THE ADDENDUM CIRCLES OF THE GEAR AND PINION
ARC OF CONTACT	PATH TRACED FROM THE BEGINNING TO THE END OF ENGAGEMENT OF A GIVEN PAIR OF TEETH
ARC OF APPROACH	PORTION OF THE PATH OF CONTACT FROM THE BEGINNING OF THE ENGAGEMENT TO THE PITCH POINT
ARC OF RECESS	PORTION OF THE PATH OF CONTACT FROM THE PITCH POINT TO THE END OF THE ENGAGEMENT OF A PAIR OF TEETH.

ANGLE OF ACTION	ANGLE FROM A TEETH COME INTO CONTACT UNTIL THEY GO OUT OF CONTACT
ARC OF ACTION	SUBTENDS THE ANGLE OF ACTION
ANGLE OF APPROACH	ANGLE FROM A TEETH COME IN TO CONTACT UNTIL CONTACT AT THE PITCH POINT
ANGLE OF RECESS	ANGLE FROM A TEETH ARE IN CONTACT AT PITCH POINT UNTIL THEY PASS OUT OF MESH
VELOCITY RATIO	ANGULAR VELOCITY OF THE DRIVER DIVIDED BY THE DRIVEN GEAR
GEAR RATIO	NUMBER OF TEETH IN GEAR OVER PINION
CONTACT RATIO	RATIO OF THE LENGTH OF ARC OF CONTACT TO CIRCULAR PITCH
BASE CIRCLE	CIRCLE WHERE THE INVOLUTE IS GENERATED
DEGREE OF INVOLUTE	DEFINE THE BASE CIRCLE FOR A PARTICULAR PITCH CIRCLE
PRESSURE ANGLE (ANGLE OF OBLIQUITY)	ANGLE BETWEEN THE COMMON NORMAL TO TWO GEAR TEETH AT THE POINT OF CONTACT AND THE COMMON TANGENT AT THE PITCH POINT. DENOTED BY φ. THE STANDARD PRESSURE ANGLES ARE **14.5° AND 20°**

GEAR PROPORTIONS

CIRCULAR PITCH AND DIAMETRAL PITCH

$$P_c = \frac{\pi D_p}{N} = \pi M \qquad P_d = \frac{N}{D} = \frac{1}{M}$$

$$P_d \times P_c = \pi$$

D_p = PITCH Ø
N = NO. OF TEETH
M = MODULE

CLEARANCE

$$W_n = a + d = W_r + c$$
$$W_r = 2a$$
$$\text{CLEARANCE} = d - a$$

W_n = WHOLE DEPTH
W_r = WORKING DEPTH
a = ADDENDUM
d = DEDENDUM

GEAR DIAMETERS

$$D_o = D_p + 2a$$
$$D_r = D_p - 2d$$
$$D_b = D_p \cos\phi$$

D_o = OUTSIDE Ø
D_r = ROOT Ø
D_b = BASE Ø

CENTER DISTANCES

EXTERNAL GEARS

$$C = \frac{1}{2}(D_g + D_p) = \frac{1}{2P_d}(N_g + N_p)$$

INTERNAL GEARS

$$C = \frac{1}{2}(D_g - D_p) = \frac{1}{2P_d}(N_g - N_p)$$

FACEWIDTH

$$2.5P_c < b < 4P_c \quad | \quad \frac{8}{P_d} < b < \frac{12.5}{P_d}$$

BACKLASH

$B = S - T$
S → TOOTH THICKNESS
T → TOOTH SPACE

$$B = \frac{0.03}{P_d} - \frac{0.04}{P_d}$$

STANDARD GEAR TOOTH PROPORTIONS

	14.5° BROWN & SHARPE, 14.5° CYCLOIDAL	14.5° FULL DEPTH	20° FULL DEPTH	20° STUB	20° FELLOWS
ADDENDUM	$\frac{1}{P_d}$	$\frac{1}{P_d}$	$\frac{1}{P_d}$	$\frac{0.8}{P_d}$	$\frac{1}{P_d}$
DEDENDUM	$\frac{1.157}{P_d}$	$\frac{1.157}{P_d}$	$\frac{1.157}{P_d}$	$\frac{1}{P_d}$	$\frac{1.25}{P_d}$
CLEARANCE	—	DEDENDUM − ADDENDUM			—
TOOTH THICKNESS	—		$\frac{1.5708}{P_d}$	—	
FILLET RADIUS	$\frac{0.209}{P_d}$	$\frac{0.209}{P_d}$	$\frac{0.236}{P_d}$	$\frac{0.3}{P_d}$	$\frac{0.25}{P_d}$

CONTACT RATIO

- RATIO OF LENGTH OF LINE OF ACTION TO BASE PITCH.

$$C.R. = \frac{\sqrt{(R_o^2 - R_b^2)_g} + \sqrt{(R_o^2 - R_b^2)_p} - c\sin\theta}{P_c \cos\theta}$$

NOTE:
N = No. OF TEETH
n = RPM

GEAR TRAINS

SPEED RATIO

$$\text{SPEED RATIO} = \left(\frac{N_2}{N_1}\right)\left(\frac{N_4}{N_3}\right)\left(\frac{N_6}{N_5}\right) = \frac{n_1}{n_6}$$

n_1 ← 1st DRIVER
n_6 ← LAST DRIVER

SAME SHAFT : 1 2 $\quad n_1 = n_2$

MESHING : 1 2 $\quad V_1 = V_2 \; ; \; D_1 n_1 = D_2 n_2$

DESIGN CALCULATIONS:

BEAM FATIGUE STRENGTH

$$F_s = \frac{(Sn)(b)(Y)}{k_f P_d}$$

Y = LEWIS FORM FACTOR
b = FACE WIDTH
Pd = DIAM. PITCH

ENDURANCE STRENGTH

Sn = ENDURANCE STRENGTH

$Sn = \frac{Su}{3}$ (VALLANCE)

$Sn = \frac{Su}{2}$ (FAIRES)

= 250 BHN (FOR STEEL)

STRESS REDUCTION FACTOR

kf = STRESS REDUC. FACTOR
= 1.45 (INTERMITTENT)
= 1.7 (CONTINUOUS)

DYNAMIC LOAD

CONTINUOUS

BUCKINGHAM'S:

$$F_d = F_t + \frac{0.05V_m(bC + F_t)}{0.05V_m + \sqrt{bC + F_t}}$$

INTERMITTENT SERVICE

APPLICATION	CONDITION	FORMULA
COMMERCIAL CUT	$V_m \leq 2000$ fpm	$F_d = \left[\frac{600+V_m}{600}\right]F_t$
CAREFULLY CUT	$1000 < V_m < 4000$ fpm	$F_d = \left[\frac{1200+V_m}{1200}\right]F_t$
PRECISION CUT	$V_m > 4000$ fpm	$F_d = \left[\frac{78+\sqrt{V_m}}{78}\right]F_t$
COMMERCIALLY HOBBED OR SHAVE TEETH	-	$F_d = \left[\frac{50+\sqrt{V_m}}{50}\right]F_t$

CHECKING FOR FAILURE BY FATIGUE

$$N_{sf} = \frac{F_s}{F_d}$$

UNIFORM LOAD W/O SHOCK	1.0 – 1.25
MEDIUM SHOCK	1.25 - 1.50
MODERATELY HEAVY SHOCK	1.50 - 1.75
HEAVY SHOCK	1.75 - 2.0

WEAR LOAD

$$F_w = D_p \times b \times Q \times k_g$$

Q = RATIO FACTOR FOR GEARING
kg = WEAR FACTOR
Dp = PINION PITCH Ø

RATIO FACTOR FOR GEARING

$$Q = \frac{2D_g}{D_g \pm D_p} = \frac{2N_g}{N_g \pm N_p} = \frac{2m_g}{m_g \pm 1}$$

NOTE:
- \+ EXTERNAL GEARING
- \- INTERNAL GEARING

MATERIAL FACTOR (WEAR FACTOR)

$$k_g = \frac{(S_n)^2 \sin\phi}{1.4}\left(\frac{1}{E_p} + \frac{1}{E_g}\right)$$

SAFE DESIGN: $F_w \geq F_d$

TRANSMITTED LOAD (Ft)

$$F_t = F_n \cos\phi$$

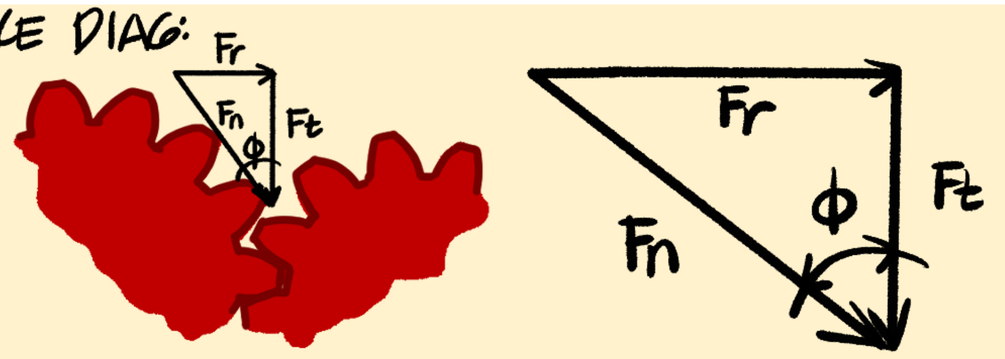
NORMAL LOAD (Fn)

$$F_n = \sqrt{F_t^2 + F_r^2}$$

SEPARATING LOAD (Fr)

$$F_r = F_n \sin\phi$$

FORCE DIAG:



CYCLOID	ROLLS ALONG A STRAIGHT LINE
EPICYCLOID	ROLLS ALONG OUTER SIDE OF ANOTHER CIRCLE
HYPOCYCLOID	ROLLS ALONG INNER SIDE OF ANOTHER CIRCLE

$$C = \frac{kE_gE_p}{E_g + E_p}$$

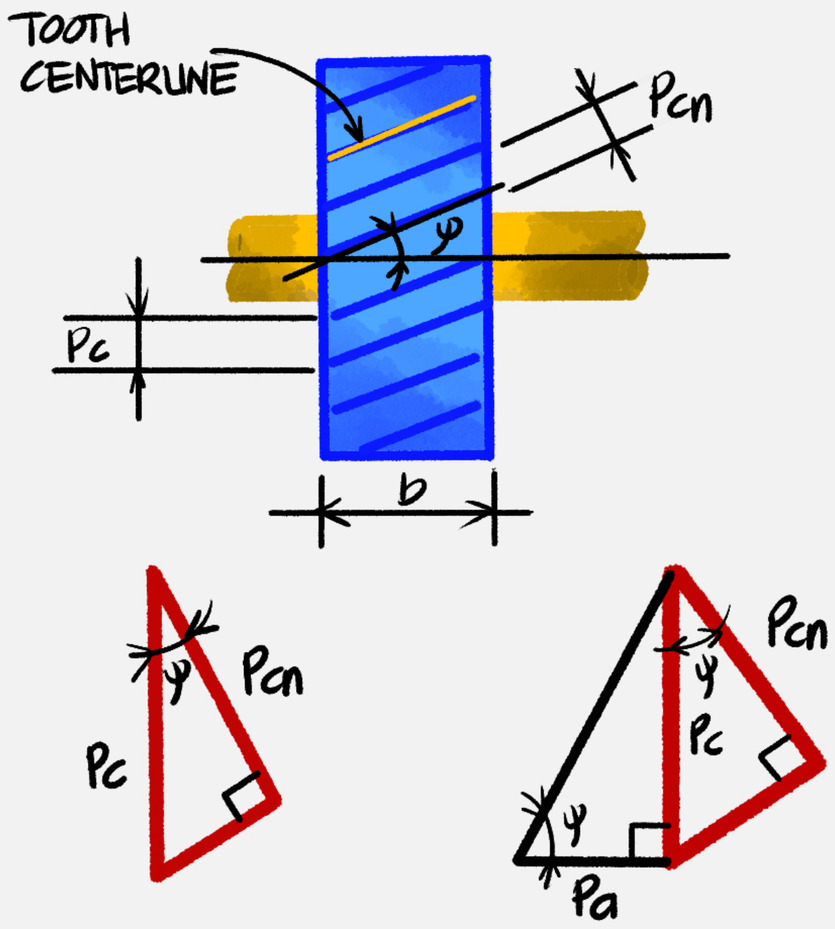
E = MODULUS OF ELASTICITY
C = FACTOR

k = 0.107e - 14.5° FD
= 0.111e - 20° FD
= 0.115e - 20° STUB

HELICAL GEARS

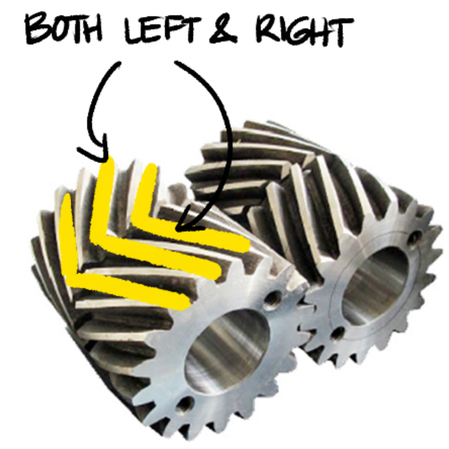
- ARE TOOTHED WHEELS WHOSE **TOOTH ELEMENTS ARE CUT IN THE FORM OF A HELIX** ABOUT AN AXIS OF ROTATION AND ARE USED TO **CONNECT PARALLEL SHAFTS.**

HELICAL GEAR: NOMENCLATURE

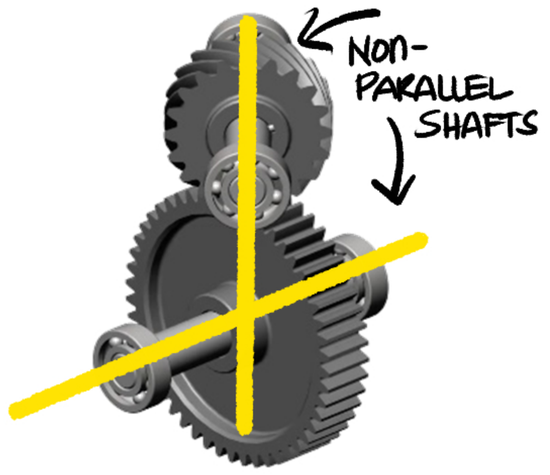


HELIX ANGLE	ANGLE BETWEEN A TANGENT TO THE PITCH HELIX AND AN INTERSECTING AXIAL ELEMENT OF THE PITCH CYLINDER
BASE HELIX ANGLE	ANGLE BETWEEN A TANGENT TO THE BASE HELIX AND AN ELEMENT OF THE BASE CYLINDER.
FACE CONTACT RATIO	IS THE ADVANCE OF THE TOOTH IN THE LINE OF FACE WIDTH DIVIDED BY THE CIRCULAR PITCH

HERRINGBONE	ARE DOUBLE HELICAL GEARS CONSISTING OF RIGHT-HAND AND LEFT-HAND HELICES TO **ABSORB THE AXIAL THRUST** WITHIN THE GEAR AND ARE USED TO **CONNECT PARALLEL SHAFTS.**
CROSSED HELICAL	HELICAL GEARS MOUNTED ON NON-PARALLEL SHAFTS

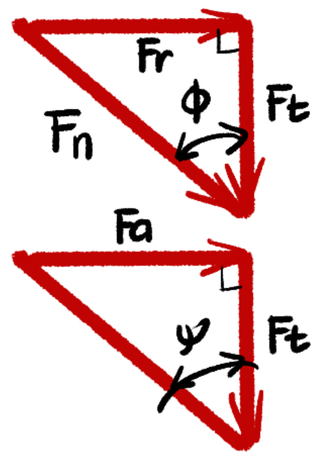


(a) HERRINGBONE GEARS



(b) CROSSED HELICAL GEARS

FORCES



$F_t = F_n \cos\phi$

$F_r = F_n \sin\phi$

(SAME AS SPUR)

@ AXIAL FORCE

$F_a = F_t \tan\psi$

GEAR PROPORTIONS

PITCHES

$\cos\psi = \frac{P_{cn}}{P_c}$ | $\tan\psi = \frac{P_c}{P_a}$

$P_{cn} = P_c \cos\psi$

$P_{cn} = \frac{\pi D}{N}\cos\psi$

$\cos\psi = \frac{\pi / P_{dn}}{\pi / P_d}$

$P_{dn} = \frac{P_d}{\cos\psi}$

$\tan\psi = \frac{P_{cn}}{\cos\psi}\,/\,P_a$

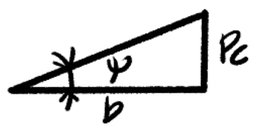
$P_a = \frac{P_{cn}}{\tan\psi\cos\psi}$

$P_a = \frac{P_{cn}}{\sin\psi}$

NOTE:
ψ = HELIX ANGLE
Pcn = NORMAL CIRCULAR PITCH
Pc = CIRCULAR PITCH
Pa = AXIAL PITCH
Pd = DIAMETRAL PITCH

FACE WIDTH

$b = \frac{P_c}{\tan\psi} \geq 2P_a$



$b = m_f P_a$

m_f = FACE CONTACT RATIO

LEAD

$L = \frac{\pi D}{\tan\psi} = nP_a$

NO. OF HELICES (n)
1 – SINGLE HELIX
2 – DOUBLE HELIX

ANGLE RELATIONSHIP

$\tan\phi_n = \tan\phi\cos\psi$

ϕ = PRESSURE ∠
ϕ_n = NORMAL PRESSURE ∠
ψ = HELIX ∠

DESIGN CALCULATIONS

FATIGUE STRENGTH

$F_s = \frac{S_b Y}{k_f P_{dn}}$

FORMATIVE NUMBER OF TEETH

$N_{ev} = \frac{N}{\cos^3\psi}$

DYNAMIC LOAD

$F_d = F_t + \frac{0.05 V_m (F_t + bC\cos^2\psi)\cos\psi}{0.05 V_m + \sqrt{F_t + bC\cos^2\psi}}$

LIMITING WEAR LOAD

$F_W = \frac{D_p b Q k_g}{\cos^2\psi}$ | $k_g = \frac{S_n^2 \sin\phi_n}{1.4}\left[\frac{1}{E_p} + \frac{1}{E_g}\right]$

CROSSED-HELICAL GEARS

SUMMATION OF ANGLES

Σ → CAN BE AT ANY ANGLE ; BUT $\Sigma = 90°$ (MOST COMMON)

- ☑ SAME HANDS : $\Sigma = \psi_1 + \psi_2$
- ☑ DIFFERENT HANDS : $\Sigma = \psi_1 - \psi_2$

GEAR PROPORTIONS

$$P_{cn} = P_c \cos\psi$$

$$P_{cn} = \frac{\pi D_1}{N_1} \cos\psi_1$$

$$P_{cn} = \frac{\pi D_2}{N_2} \cos\psi_2$$

SPEED RATIO

$$m_W = \frac{n_1}{n_2} = \frac{\omega_1}{\omega_2}$$

$$m_W = \frac{N_2}{N_1}$$

$$m_W = \frac{D_2 \cos\psi_2}{D_1 \cos\psi_1}$$

GEAR PROPORTIONS (FOR 90 DEG.)

FOR $\Sigma = 90°$

$\psi_1 + \psi_2 = 90°$

$\cos\psi_1 = \cos(90 - \psi_2)$

- ➤ $\cos\psi_1 = \sin\psi_2$
- ➤ $\cos\psi_2 = \sin\psi_1$

SPEED RATIO:

$$m_W = \frac{n_1}{n_2} = \frac{\omega_1}{\omega_2}$$

$$m_W = \frac{N_2}{N_1}$$

$$m_W = \frac{D_2 \cos\psi_2}{D_1 \cos\psi_1} = \frac{D_2 \tan\psi_1}{D_1}$$

EFFICIENCY

$$e = \frac{\cos\phi_n - f\cot\psi_1}{\cos\phi_n + f\tan\psi_1}$$

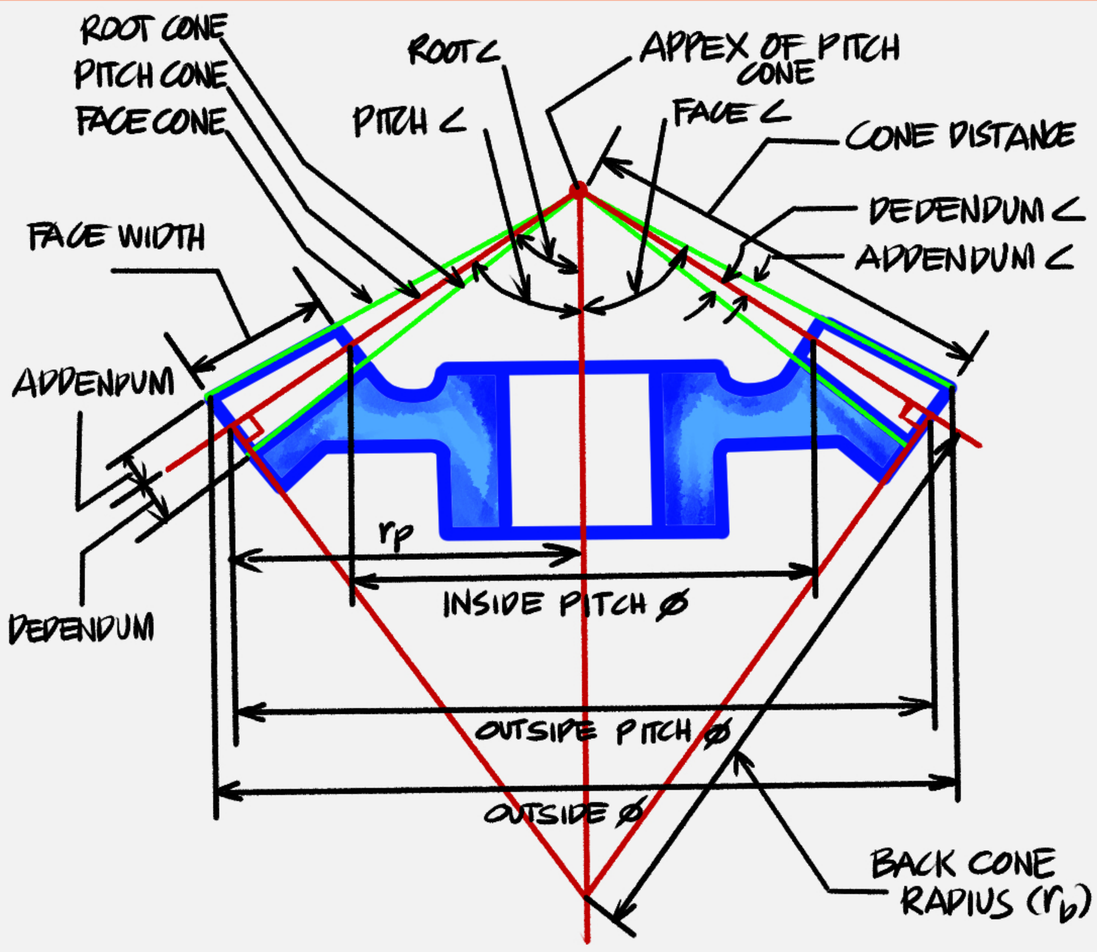
ϕ_n = NORMAL PRESSURE ∠

f = COEF. OF FRICTION

BEVEL GEARS

- BEVEL GEARS HAVE TEETH FORMED ON A CONICAL SURFACE AND ARE USED TO TRANSMIT MOTION BETWEEN NONPARALLEL SHAFTS.

BEVEL GEAR: NOMENCLATURE



TYPES OF BEVEL GEARS

STRAIGHT BEVEL	TEETH ARE STRAIGHT BUT WITH TAPERED SIDES
SPIRAL BEVEL GEARS	CURVED OBLIQUE TEETH, CONTACT BEGINS GRADUALLY AND CONTINUES SMOOTHLY FROM END TO END. WITH SPIRAL ANGLES. FOR HIGHER SPEEDS
ZEROL BEVEL GEARS	TEETH ARE CURVED BUT LIE IN THE SAME DIRECTION AS THE TEETH OF STRAIGHT BEVEL GEARS. ZERO SPIRAL ANGLE
HYPOID GEARS	RESEMBLES SPIRAL BEVEL GEARS EXCEPT THAT THE SHAFT AXES ARE OFFSET (NOT INTERSECTING), THE **PITCH SURFACE IS A HYPERBOLOID.**
SPHEROID GEARS	SIMILAR TO HYPOID BUT WITH **LARGER** OFFSET

GEAR CLASSIFICATION ACCDG. TO PITCH ANGLES

MITER GEARS	SHAFTS AT 90°, 45° EACH
CROWN GEAR	90° PITCH ANGLE
INTERNAL BEVEL	PITCH ANGLE > 90°
EXTERNAL BEVEL	PITCH ANGLE < 90°

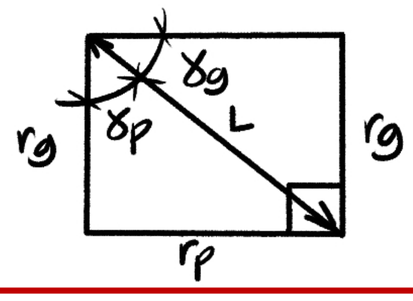
PROPORTIONS OF BEVEL GEAR (GEAR RATIO OF 1)

$$\text{ADDENDUM } (a) = \frac{1}{Pd} \qquad \text{WORKING DEPTH } (Wr) = \frac{2}{Pd}$$

$$\text{DEDENDUM } (d) = \frac{1.2}{Pd} \qquad \text{WHOLE DEPTH } (Wh) = \frac{2.2}{Pd}$$

$$\text{CLEARANCE } (c) = \frac{0.2}{Pd} \qquad \text{TOOTH THICKNESS} = \frac{1.5708}{Pd}$$

PROPORTIONS WHEN SHAFT ANGLE IS 90 DEG.



$$\tan\gamma_p = \frac{r_p}{r_g} \qquad \tan\gamma_g = \frac{r_g}{r_p} \qquad L = \sqrt{r_p^2 + r_g^2}$$

PITCH ANGLE

$\Sigma < 90°$ (IF $\Sigma = 90°$; $\sin 90° = 1$ & $\cos 90° = 0$)

$$\tan\gamma_g = \frac{\sin\Sigma}{\frac{r_p}{r_g} + \cos\Sigma} \qquad \tan\gamma_p = \frac{\sin\Sigma}{\frac{r_g}{r_p} + \cos\Sigma}$$

FACE WIDTH

FOR STRAIGHT/SPIRAL

$b \le \frac{L}{3}$ or $\frac{10}{Pd}$

USE → $b = \frac{L}{3}$

FOR ZEROL

$b \le 0.275L$ or $\frac{10}{Pd}$

USE → $b = 0.25L$

FORMATIVE NUMBER OF TEETH

$$N_{ep} = \frac{2 r_b P_d}{} = \frac{2 r_p L P_d}{r_g} \rightarrow \text{FOR PINION}$$

$$N_{ep} = \frac{2 r_g L P_d}{r_p} \rightarrow \text{FOR GEAR}$$

$$\frac{r_b}{L} = \frac{r_p}{r_g}$$

$$F_a = F_t \tan\phi \sin\gamma \qquad F_r = F_t \tan\phi \cos\gamma$$

FATIGUE STRENGTH OF STRAIGHT BEVEL GEARS

$k_f \rightarrow$ NEGLIGIBLE

$$F_s = \frac{S Y b}{P_d}\left[1 - \frac{b}{L} + \frac{b^2}{3L^2}\right]$$

FATIGUE STRENGTH OF STRAIGHT BEVEL GEARS

WHEN $b \le L/3$

$$F_s = \frac{S Y b}{P_d}\left[\frac{L - b}{L}\right]$$

RATED STRENGTH OF BEVEL GEARS

$k_t = \frac{460 + °F}{620}$

$$F_s = \frac{S b J}{P_d}\left(\frac{k_L}{k_s k_t k_r}\right)$$

J = GEOMETRY FACTOR
k_L = LIFE FACTOR k_t = TEMP. FACTOR
k_s = SIZE FACTOR k_r = RELIABILITY FACTOR

DYNAMIC LOAD

$$F_d = (VF) N_{sf} \times k_m \times F_t$$

k_m = LOAD DISTRIBUTION FACTOR

BOTH GEARS STRADDLE MOUNTED	1–1.1
ONE GEAR STRADDLE, ONE NOT	1.1–1.25
BOTH GEARS OVERHUNG	1.25–1.4

STRAIGHT

$$VF = \frac{50 + \sqrt{V_m}}{50}$$

SPIRAL

$$VF = \left[\frac{78 + \sqrt{V_m}}{78}\right]^{1/2}$$

NOTE: $VF = 1 \rightarrow$ DYNAMIC INCREMENT NEGLIGIBLE

RATED WEAR LOAD FOR BEVEL GEARS

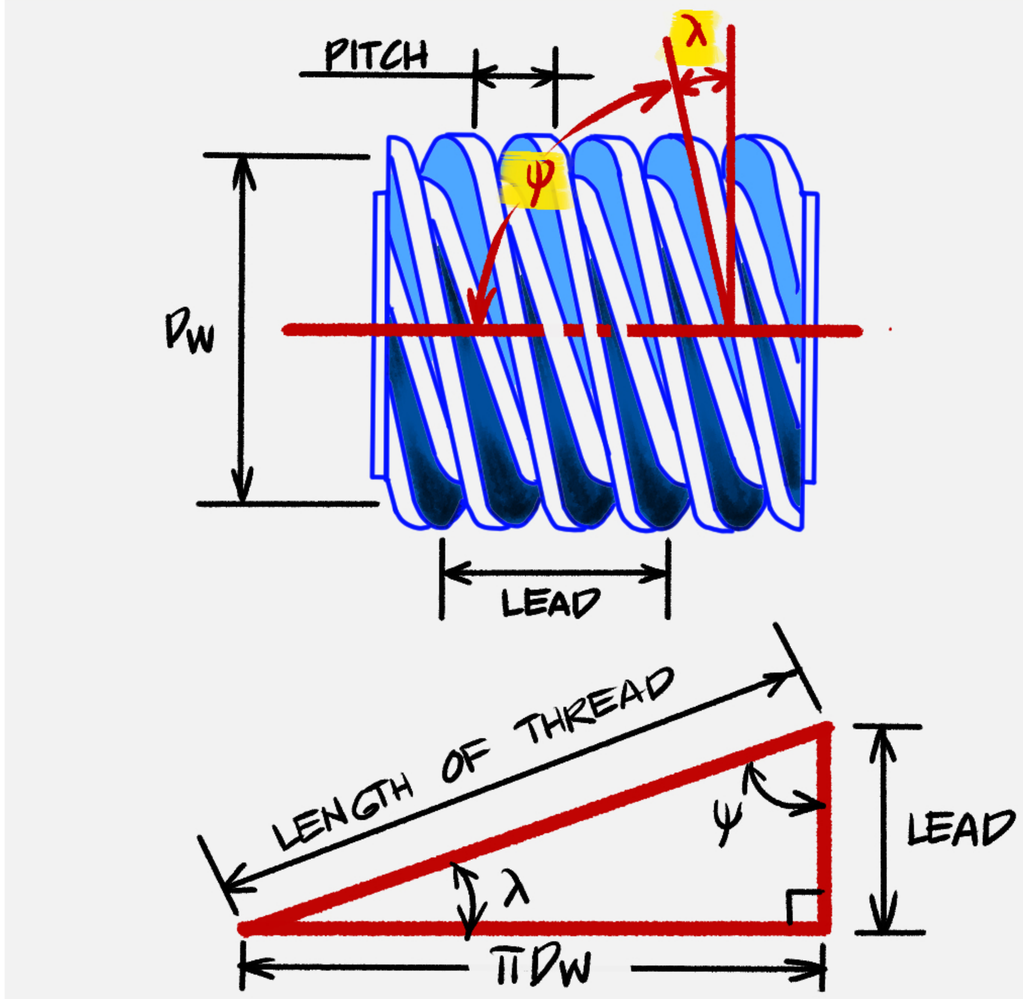
$$F_w = D_p b I \times \frac{S_{cd}^2}{C_e^2} \times \left(\frac{C_L}{K_t C_r}\right)^2$$

S_{cd} = DESIGN CONTACT STRESS

WORM GEARS

- ARE WIDELY USED FOR TRANSMITTING POWER AT HIGH VELOCITY RATIOS BETWEEN NON-INTERSECTING SHAFTS THAT ARE GENERALLY, BUT NOT NECESSARILY, AT RIGHT ANGLES.

WORM GEAR: NOMENCLATURE



AXIAL PITCH (LINEAR PITCH OF WORM) — DISTANCE MEASURED AXIALLY (I.E. PARALLEL TO THE AXIS OF WORM) FROM A POINT ON ONE THREAD TO THE ADJACENT THREAD

LEAD — LINEAR DISTANCE THAT A POINT ON A THREAD MOVES IN ONE REVOLUTION OF THE WORM.

LEAD ANGLE — ANGLE BETWEEN THE TANGENT TO THE THREAD HELIX AND THE PLANE NORMAL TO THE AXIS OF THE WORM.

TOOTH PRESSURE ANGLE — MEASURED IN A PLANE CONTAINING THE AXIS OF THE WORM AND IS EQUAL TO ONE-HALF THE THREAD PROFILE ANGLE.

NORMAL PITCH — DISTANCE ALONG THE NORMAL TO THE THREADS BETWEEN TWO POINTS ON ADJACENT THREADS

HELIX ANGLE — ANGLE BETWEEN THE TANGENT TO THE THREAD HELIX AND THE AXIS OF THE WORM.

VELOCITY RATIO — RATIO OF THE SPEED OF WORM TO WORM GEAR

GEAR PROPORTIONS

$\phi_n = 14.5°$; $\lambda = 16°$
$\phi_n = 20°$; $\lambda = 25°$
$\phi_n = 25°$; $\lambda = 35°$
$\phi_n = 30°$; $\lambda = 45°$

PITCHES AND LEAD ANGLE

$$P_{cn} = P_c \cos\psi$$
$$P_{cn} = P_a \cos\psi$$
$$P_{a_W} = P_{c_g} \quad [\text{WHEN } \Sigma = 90°]$$
$$P_{dn} = \frac{P_d}{\cos\psi}$$

☑ LEAD ANGLE, λ

$$\tan\lambda = \frac{\text{LEAD}}{\pi D_W}$$

☑ LEAD

$$L = N_t P_a$$

N_t → NUMBER OF THREADS

$$\lambda + \psi_W = 90°$$
$$\lambda_g + \psi = 90°$$

➢ IF SHAFT $\angle = 90°$

$$\lambda = \psi \rightarrow \lambda_g = \psi_W$$

FACE WIDTH AND DIAMETERS

$$b = \frac{1}{2} D_{W_o}$$
$$D_{W_o} = D_W + 2a$$

$$D_W = \frac{C^{0.875}}{2.2}$$

C = CENTER DISTANCE

$$a = 0.3183 P_{cn}$$

VELOCITY RATIO

$$m_W = \frac{D_g \cos\psi_g}{D_W \cos\psi_W}$$

IF SHAFT $\angle = 90°$
$\psi_g = \lambda_W$
$\lambda_W + \psi_W = 90°$

$$\rightarrow m_W = \frac{D_g \cos\lambda}{D_W \sin\lambda}$$

TORQUE TRANSMITTED BY WORM

$$T = W_T \left[D_W / 2 \right]$$

TANGENTIAL FORCE

$$W_T = F_t \left[\frac{\cos\phi_n \sin\lambda + f\cos\lambda}{\cos\phi_n \cos\lambda - f\sin\lambda} \right]$$

SEPARATING FORCE

$$S = F_t \left[\frac{\sin\phi_n}{\cos\phi_n \cos\lambda - f\sin\lambda} \right]$$

EFFICIENCY

$$e = \tan\lambda \left[\frac{\cos\phi_n - f\tan\lambda}{\cos\phi_n \tan\lambda + f} \right]$$

COEFFICIENT OF FRICTION

☑ FOR $3 < V_r < 70$ fpm

$$f = \frac{0.155}{V_r^{0.2}}$$

☑ FOR $70 < V_r < 3000$ fpm

$$f = \frac{0.32}{V_r^{0.36}}$$

RUBBING SPEED

$$V_r = \frac{\pi D_W n_W}{12 \cos\lambda}$$

D_W = WORM Ø, INCHES
n_W = RPM OF WORM

NORMAL PRESSURE

$$\tan\phi_n = \tan\phi \cos\lambda$$

➢ SELF-LOCKING: $f > \tan\lambda$

DESIGN CALCULATIONS

BEAM FATIGUE

$$F_s = \frac{SYb}{P_{dn}} = \frac{SYb P_{cn}}{\pi}$$

DYNAMIC LOAD

$$F_d = \left[\frac{1200 + V_{mg}}{1200} \right] F_t$$

WEAR LOAD

$$F_w = D_g \times b \times k_w$$

THERMAL CAPACITY

$$Q = (1-e)(hp)$$
$$Q_c = h_{cr} A \Delta t$$

MINIMUM AREA (HOUSING)

$$A_{MIN} = 43.2\, C^{1.7} \text{ ; in}^2$$

➢ LUBRICANT TEMP ≤ 190°F

BEARINGS

- MACHINE ELEMENT THAT **SUPPORTS ANOTHER MOVING MACHINE ELEMENT**. THIS ELEMENT PERMITS A RELATIVE MOTION BETWEEN THE CONTACT SURFACES OF THE MEMBERS, WHILE CARRYING THE LOAD.

CLASSIFICATION OF BEARINGS

DEPENDING UPON THE DIRECTION OF LOAD TO BE SUPPORTED	
RADIAL BEARINGS	THE LOAD ACTS **PERPENDICULAR TO THE DIRECTION OF MOTION** OF THE MOVING ELEMENT
THRUST BEARINGS	THE LOAD ACTS **ALONG THE AXIS OF ROTATION,** DEPENDING UPON THE NATURE OF CONTACT
DEPENDING UPON THE NATURE OF CONTACT	
SLIDING CONTACT BEARINGS	THE SLIDING TAKES PLACE ALONG THE **SURFACES OF CONTACT** BETWEEN THE MOVING ELEMENT AND THE FIXED ELEMENT
ROLLING CONTACT BEARINGS	WHERE THE **STEEL BALLS OR ROLLERS ARE INTERPOSED** BETWEEN THE MOVING AND FIXED ELEMENTS, OFFERS ROLLING FRICTION ACTION

TYPES OF SLIDING CONTACT BEARINGS

JOURNAL BEARINGS	ARE USED TO FURNISH LATERAL SUPPORT TO ROTATING SHAFTS. THE JOURNAL IS THE PART OF THE SHAFT THAT RUNS IN THE BUSHING OR SLEEVE
FULL JOURNAL BEARING	WHERE THE ANGLE OF CONTACT OF THE BUSHING WITH THE JOURNAL IS **360**
PARTIAL JOURNAL BEARING	WHERE THE ANGLE OF CONTACT OF BUSHING THE JOURNAL IS **180** OR LESS, **120** BEING A COMMON VALUE
CLEARANCE BEARING	REFERS TO THE THICKNESS OF THE SPACE ALLOWED FOR THE LUBRICANT THAT SEPARATES THE PARTS HAVING RELATIVE MOTION
FITTED BEARING	IS ONE IN WHICH THE RADIUS OF THE JOURNAL AND BUSHING ARE EQUAL
THRUST BEARING	A BEARING DESIGNED TO SUPPORT AN AXIAL LOAD
• **STEP BEARING**	IN WHICH THE END OF THE SHAFT IS IN CONTACT WITH A BEARING SURFACE
• **COLLAR BEARING**	IN WHICH A COLLAR IS ATTACHED TO OR FORMED INTEGRAL WITH THE SHAFT
• **PIVOTED-SEGMENT BEARING**	WAS DEVELOPED IN ORDER TO PROVIDE A CONVERGING FILM IN A THRUST BEARING

PROPERTIES OF LUBRICANTS

- PROVIDE LOW-FRICTION FILM BETWEEN MATERIALS
- PROTECT BEARING FROM CORROSION
- HELP DISSIPATE HEAT

KINEMATIC VISCOSITY

$$\mathcal{M}_K = 0.22\,SSU - \frac{180}{SSU}$$

$\mathcal{M}_K$ = CENTISTOKES
SSU = SAYBOLT SECOND UNIV.

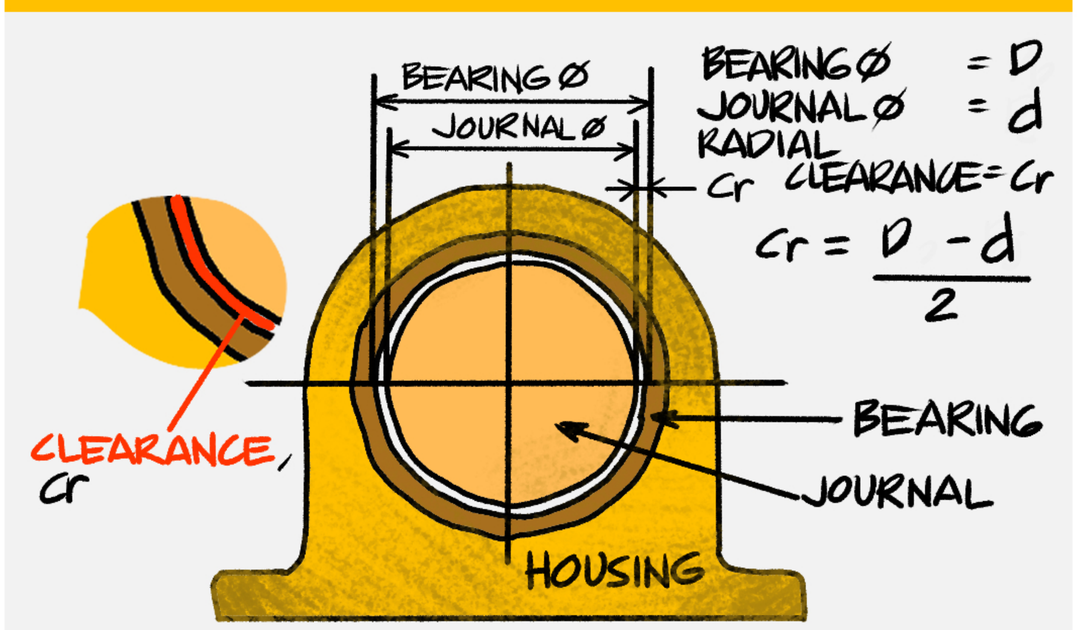
CONVERSION

$$1\ reyn = \frac{1\ lb \cdot sec}{in^2}$$

$$1\ poise = \frac{1\ dyne \cdot sec}{cm^2}$$

$1\ reyn = 6.9 \times 10^6$ CENTIPOISES

SLIDING/JOURNAL BEARINGS



DIAMETRAL CLEARANCE

$$cd = 2 \times Cr = D - d$$

RADIAL CLEARANCE

$$Cr = \frac{cd}{2}$$

DIAMETRAL CLEARANCE RATIO

RATIO OF cd TO JOURNAL Ø
$= cd/d$

ECCENTRICITY

$$e = \frac{cd}{2} - ho = Cr - ho$$

RUNNING CLEARANCE

$rc = cd - ho$ | ho = min. oil film thck.

ATTITUDE OR ECCENTRICITY RATIO

$$\mathcal{E} = \frac{Cr - h_o}{Cr}$$

MINIMUM OIL THICKNESS

$ho \approx$ ASSUMED "$\frac{cd}{4}$"

PRESSURE BETWEEN BEARING AND THE JOURNAL

$$P = \frac{\text{FORCE}}{\text{AREA}}$$ | A = PROJECTED AREA = D×L

BEARING MODULUS

$$BM = \frac{Mn}{P}$$ | M = VISCOSITY IN REYNS, n = ROTATIVE SPEED IN RPS

1 REYN = 1 lb-sec/in²
1 POISE = 1 dyne-sec/cm²

SOMMERFELD NUMBER OR BEARING CHARACTERISTICS NO.

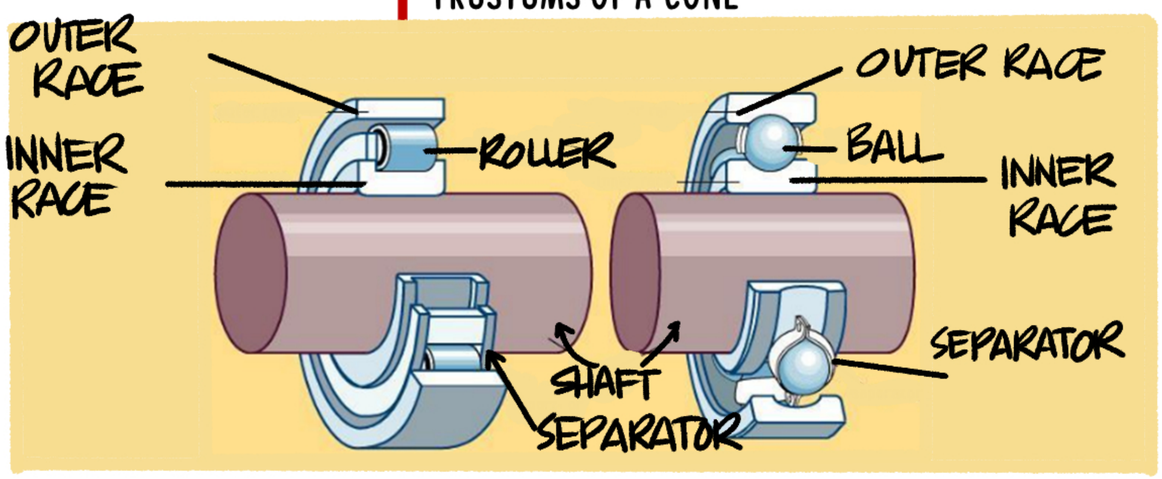
$$S = \frac{Mn_s}{P}\left(\frac{D}{cd}\right)^2 = \frac{Mn_s}{P}\left(\frac{R}{cr}\right)^2$$

TYPES OF ROLLING CONTACT BEARINGS

BALL BEARING	CONSIST OF AN INNER RACE WHICH IS MOUNTED ON THE SHAFT OR JOURNAL AND AN OUTER RACE WHICH IS CARRIED BY THE HOUSING OR CASING
ROLLER BEARING	

TYPES OF RADIAL BALL BEARINGS

DEEP-GROOVE BALL BEARING	THE BALLS ARE ASSEMBLED BY THE ECCENTRIC DISPLACEMENT OF THE INNER RING
FILLING-SLOT TYPE	HAS SLOTS OR NOTCHES THAT PERMIT THE ASSEMBLY OF MORE BALLS, GIVING A BEARING OF LARGER RADIAL LOAD CAPACITY
SELF-ALIGNING	COMPENSATE FOR ANGULAR MISALIGNMENT THAT ARISE FROM SHAFT OR FOUNDATION DEFLECTION OR ERRORS IN MOUNTING
ANGULAR-CONTACT	THE LINE THROUGH THE AREAS CARRYING THE LOAD MAKES AN ANGLE WITH THE PLANE OF THE FACE OF THE BEARING
DOUBLE-ROW	SIMILAR TO SINGLE-ROW BALL BEARINGS, EXCEPT THAT EACH RINGS HAS TWO GROOVES
CYLINDRICAL ROLLER	LINE CONTACT INSTEAD OF POINT CONTACT, WHICH RESULTS IN A GREATER AREA CARRYING THE LOAD
SELF-ALIGNING ROLLER	RUNNING IN A DOUBLE-GROOVED INNER RING, CURVED OUTER RINGS THAT LOOK LIKE THE A SELF-ALIGNING BALL BEARING
NEEDLE BEARINGS	CYLINDRICAL BEARINGS MADE WITH RELATIVELY LONG BEARINGS
TAPERED ROLLER	WHERE THE ROLLING ELEMENTS ARE FRUSTUMS OF A CONE



FRICTIONAL TORQUE

$$T_b = F \times f_b \times \frac{D}{2}$$ | f_b = COEFFICIENT OF FRICTION

PETROFF'S EQUATION FOR FRICTIONAL TORQUE

$$T_f = \frac{4M\pi^2 r^3 L n_s}{c_r}$$ | T_f = lb.in; M = reyns; r = journal radius; in.; L, c_r } in; n_s = rps

FRICTIONAL HORSEPOWER

$$P = 2\pi T_f n \quad \text{or} \quad P = fWV_m$$ | W = LOAD ON EACH BEARING

HEAT DISSIPATION IN JOURNAL

$$H = C_n LD/778 \;;\; \frac{BTU}{min}$$ | C_n = HEAT DISSIPATION COEFF.

USUAL JOURNAL BEARING RATIO

L/D = 1
L/D = 0.5
L/D = 0.25

L/D > 1 LONG BEARING
L/D < 1 SHORT BEARING

FORCES, STRESSES, AND POWER

$$P = \frac{F}{A} = \frac{F}{L \times D}$$
$$T_f = fF \times \frac{D}{2}$$
POWER = fWV = 2πTN
$S_{SMAX} = 0.315 S_{MAX}$
S_{SMAX} = MAX. CONTACT STRESS
S_{MAX} = MAX. HERZIAN CONTACT STRESS

LIFE IN MILLION OF REVOLUTIONS

FOR BALL BEARINGS	$L_3 = \left(\frac{C}{P}\right)^3$
FOR ROLLER BEARINGS	$L_{10} = \left(\frac{C}{P}\right)^{10/3}$

LIFE IN HOURS

FOR BALL BEARINGS	$L_3 = \frac{16{,}667}{rpm}\left(\frac{C}{P}\right)^3$
FOR ROLLER BEARINGS	$L_{10} = \frac{16{,}667}{rpm}\left(\frac{C}{P}\right)^{\frac{10}{3}}$

C = SPEC. DYNAMIC CAP. }lb.
P = EQUIVALENT LOAD

COMPRESSIVE BREAKING LOAD

$$F_c = \frac{k}{FS} D^2 \;;\; lb$$

k = 100,000 CARBON STEEL
= 125,000 HARDENED ALLOY STEEL
F = SAFETY FACTOR, 10

MAXIMUM LOAD

$$F_{MAX} = \frac{4.37}{n} F$$
F = TOTAL RADIAL LOAD

STATIC LOAD CAPACITY

$$F_s = C_s N_b D_b^2$$ | C_s = CONSTANT DEPENDING ON THE TYPE OF BEARING & MAT'LS; N_b = NUMBER OF BALLS OR ROLLERS; D_b = BALL ø, in.

DYNAMIC LOAD CAPACITY

$$Fd = C N_b^{2/3} (Nr \cos\alpha)^{0.7} D^{1.8} \;;\; lb$$ | Nr = NO. OF ROWS OF BALLS; C = CONSTANT THAT VARIES TYPE OF BALL BEARING; α = LOCATES THE PLANE

EQUIVALENT DYNAMIC LOAD

☐ $F_z/CrF_x \leq Q$
$Fe = CrF_x$
☐ $F_z/CrF_x > Q$
$Fe = 0.56CrF_x + CtF_z$

F_x = RADIAL LOAD; F_z = THRUST LOAD; Q = TABLE 12.2 of FAIRIES
Cr = ROTATION FACTOR
Cr = 1 INNER RACE ROTATING
Cr = 1.2 OUTER RACE ROTATING
Ct = THRUST FACTOR

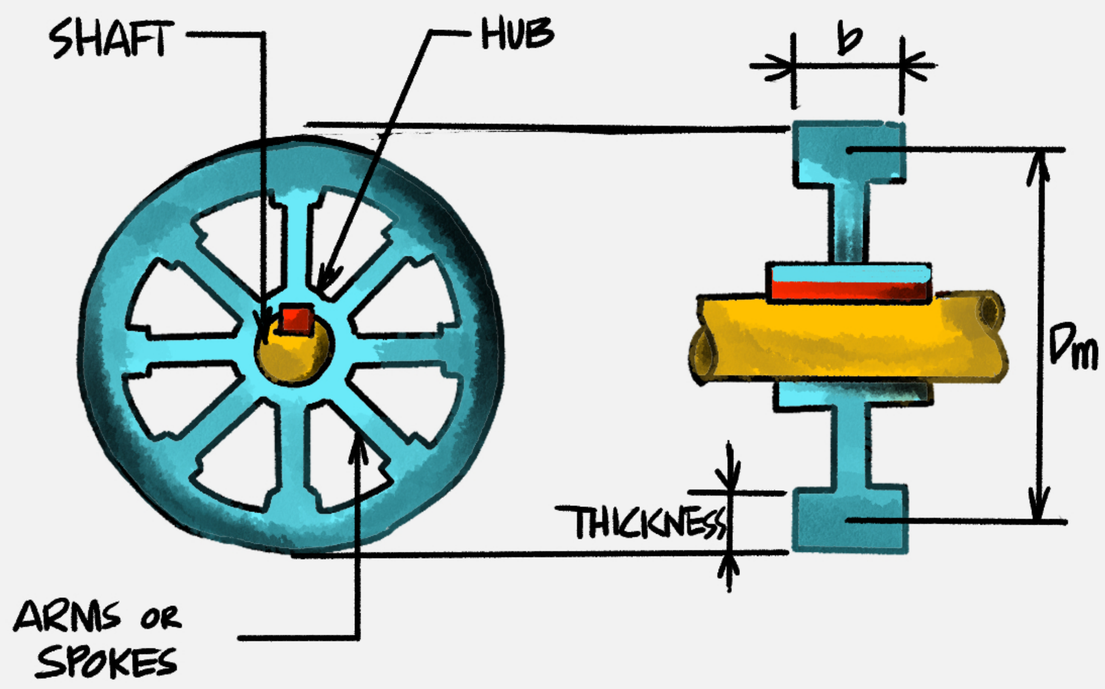
BASIC DYNAMIC LOAD RATING

$$F_r = (B_{10})^{1/3} Fe$$ | B_{10} IS THE DESIRED NUMBER OF REVOLUTIONS BEFORE 10% FAILURES HAVE OCCURED

FLYWHEELS

SERVES AS A *RESERVOIR* WHICH **STORES ENERGY** DURING THE PERIOD WHEN THE SUPPLY OF ENERGY IS MORE THAN THE REQUIREMENT AND RELEASES IT DURING THE PERIOD WHEN THE REQUIREMENT OF ENERGY IS MORE THAN SUPPLY

PARTS OF THE FLYWHEEL



TOTAL WEIGHT OF FLYWHEEL

$$W_{TOT} = W_{RIM} + W_{ARMS} + W_{HUB}$$

IF NOT SPECIFIED: $W_{AH} = 0.1 W_R$

WEIGHT OF RIM

$W_R = \rho \, Vol.R$

VOLUME OF RIM $= \pi D_m b t$

ρ = DENSITY OF FLYWHEEL
D_m = MEAN DIAMETER
b = WIDTH
t = THICKNESS

DENSITY

$= 72{,}000 \frac{kg}{m^3}$ → CAST IRON | $= 7{,}860 \frac{kg}{m^3}$ → STEEL

HOOP STRESS

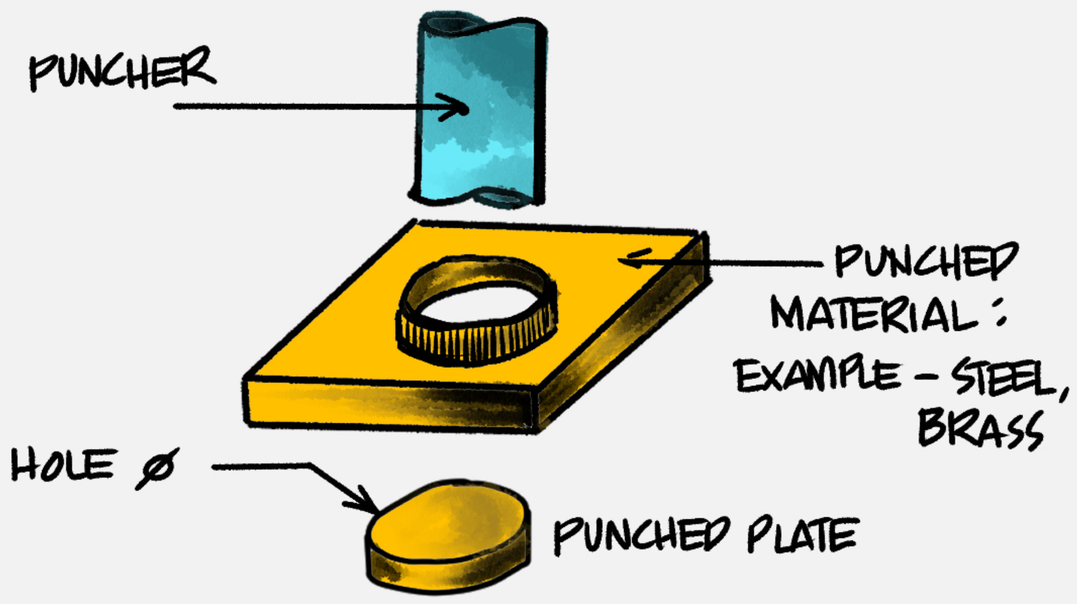
$$S = \frac{\rho}{g} \times V^2$$

ρ = DENSITY; kg/m^3, lb/in^3
V = SPEED
g = GRAVITATIONAL ACCEL.

RULE OF THUMB (CONVENTIONAL LIMITS OF OPERATION)

- CAST IRON: 6,000 fpm
- CAST STEEL: 10,000 fpm

PUNCHER



KINETIC ENERGY RELEASED BY FLYWHEEL

$$KE = \frac{1}{2} m (V_1^2 - V_2^2) = \frac{1}{2} \frac{W}{g} (V_1^2 - V_2^2) = \frac{W}{g} \times C_f V^2$$

KINETIC ENERGY IN TERMS OF MOMENT OF INERTIA

$KE = \frac{1}{2} I \omega^2$

$KE = \frac{1}{2} m k^2 (\omega_1^2 - \omega_2^2)$

I = MOMENT OF INERTIA
K = RADIUS OF GYRATION
$I = mk^2$

COEFFICIENT OF FLUCTUATION

$$C_f = \frac{V_1 - V_2}{V} \qquad C_f = \frac{\omega_1 - \omega_2}{\omega} \qquad C_f = \frac{n_1 - n_2}{n}$$

$$V = \frac{V_1 + V_2}{2} \qquad \omega = \frac{\omega_1 + \omega_2}{2} \qquad n = \frac{n_1 + n_2}{2}$$

TABLE FOR THE COEFFICIENT OF FLUCTUATION

DC GENERATOR, DIRECT DRIVE	0.002
AC GENERATOR, DIRECT DRIVE	0.035
PUNCHING SHEARING, PRESSING MACHINE	0.05 – 0.10
STAMP MILLS, CRUSHERS	0.20
RECIPROCATING PUMPS COMPRESSORS	0.03 – 0.05
MACHINE TOOLS, LOOMS, PAPER MILLS	0.025
SPINNING MILLS, FINE TO COARSE THREAD	0.01 – 0.02
GEAR DRIVES	0.02

PUNCHING STRESS

$$S_s = \frac{F}{A_s} = \frac{F}{\pi D t}$$

ENERGY NEEDED TO PUNCH A HOLE

$$E = \frac{1}{2} F \times t = \frac{1}{2} (S_s \times \pi D \times t^2)$$

NO: DO NOT USE FACTOR OF SAFETY

POWER

POWER = ENERGY / TIME

PRESSURE (IN TONS) IN PUNCHING A HOLE

FOR STEEL

- $F = 80 d \times t$

FOR BRASS

- $F = 65 d \times t$

IF NON-CIRCULAR HOLE

$d = \frac{1}{3}$ (PERIMETER)

d = DIAMETER OF HOLE
t = THICKNESS OF HOLE

SPRINGS

IS DEFINED AS AN ELASTIC BODY, WHOSE FUNCTION IS TO DISTORT WHEN LOADED AND TO **RECOVER ITS ORIGINAL SHAPE WHEN THE LOAD IS REMOVED.**

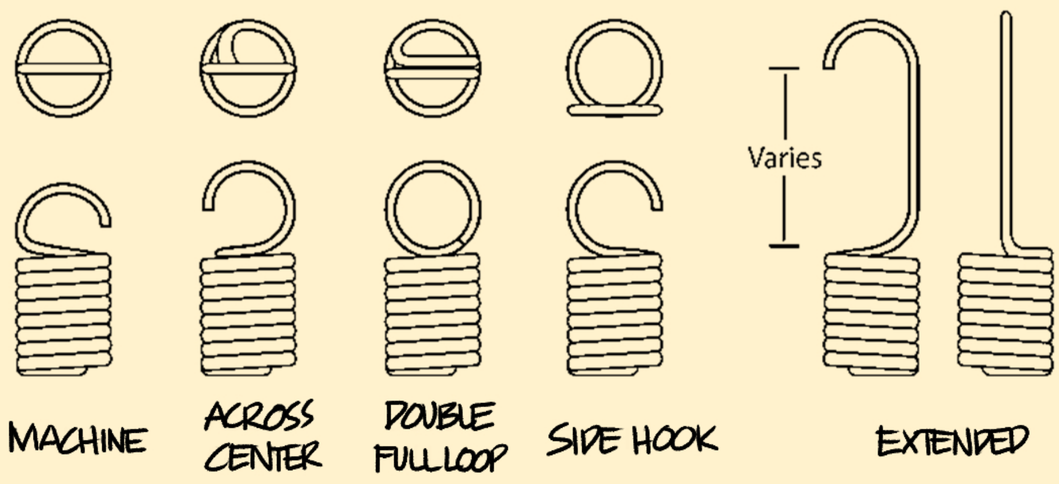
USES OF SPRINGS

- **CUSHION/ABSORB/CONTROL ENERGY** DUE TO SHOCK OR VIBRATION
- **APPLY FORCES** IN BRAKES/CLUTCHES/SPRING-LOADED VALVES.
- **CONTROL MOTION BY MAINTAINING CONTACT** BETWEEN TWO ELEMENTS (CAMS AND FOLLOWERS)
- **MEASURE FORCES** (SPRING BALANCES & ENGINE INDICATORS)
- **STORE ENERGY** (WATCHES AND TOYS)

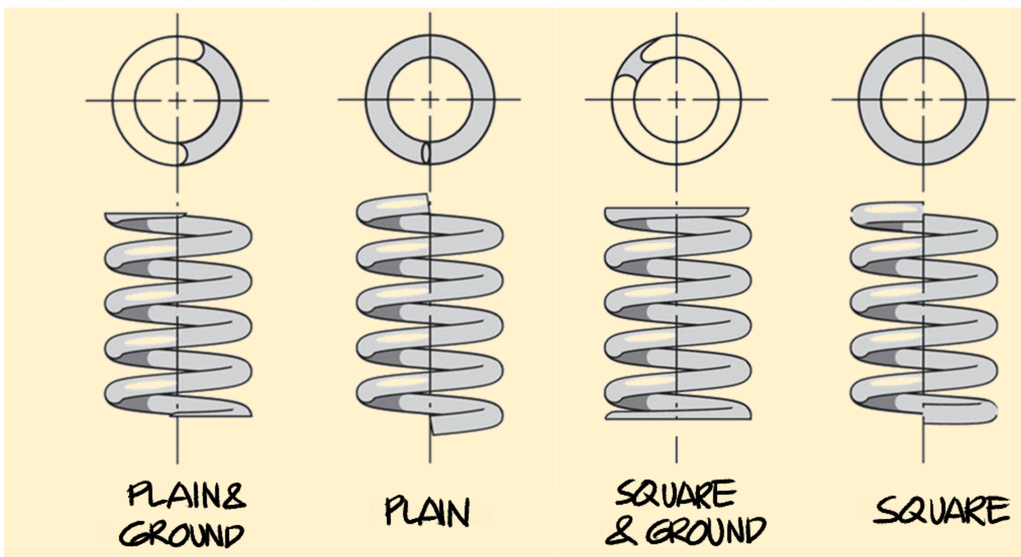
HELICAL SPRINGS

WIRE COILED IN THE FORM OF A HELIX AND IS INTENDED FOR **COMPRESSIVE OR TENSILE LOADS**. CROSS-SECTION OF THE WIRE MAY BE **CIRCULAR, SQUARE OR RECTANGULAR**.

END CONNECTION OF TENSION SPRINGS



END CONNECTION OF COMPRESSION SPRINGS



TYPES OF COILS	ACTUAL NO. OF COILS	SOLID LENGTH	FREE LENGTH
GROUND END	n	nd	np
PLAIN END	n	$(n+1)d$	$np+d$
SQUARED & GROUND END	$n+2$	$(n+2)d$	$np+2d$
SQUARED END	$n+2$	$(n+3)d$	$np+3d$

d = WIRE Ø ; p = PITCH

STRESS IN COIL SPRINGS FOR *ROUND WIRES*

$$S = \frac{8kFD_m}{\pi d^3}$$

D_m = MEAN DIAMETER
F = APPLIED FORCE
K = WHAL'S FACTOR

SQUARE WIRES

$$S = \frac{2.4kFD_m}{b^3}$$

RECTANGULAR WIRES

$$S = \frac{KFD_m(3b+1.8t)}{2b^2t^2}$$

t = THICKNESS b = WIDTH

WHAL FACTOR

☑ WHAL'S FACTOR

$$K = \frac{4c-1}{4c-4} + \frac{0.615}{c}$$

$$K_s = 1 + \frac{1}{2c} \text{ (CONCENTRATION)}$$

$$K_B = \frac{4c+2}{4c-3} \text{ (BERGTRASSER)}$$

SPRING INDEX

$$c = \frac{D_m}{d} = \frac{D_o - d}{d}$$

D_m = MEAN DIAMETER
D_o = OUTSIDE DIAMETER
$D_m = D_o - d$

SPRING RATE

$$K = \frac{F}{\delta} = \frac{F_2 - F_1}{\delta_2 - \delta_1}$$

δ = DEFLECTION

DEFLECTION IN COIL SPRINGS FOR *ROUND WIRES*

$$\delta = \frac{8FD_m n}{d^4 G} = \frac{8Fc^3 n}{dG}$$

n = effective or actual no. of coils

G = MODULUS OF RIGIDITY

SQUARE WIRES

$$\delta = \frac{2.5FD_m^3 n}{Gt^3(b-0.56)}$$

SQUARE WIRES

$$\delta = \frac{5.575FD_m^3 n}{b^4 G}$$

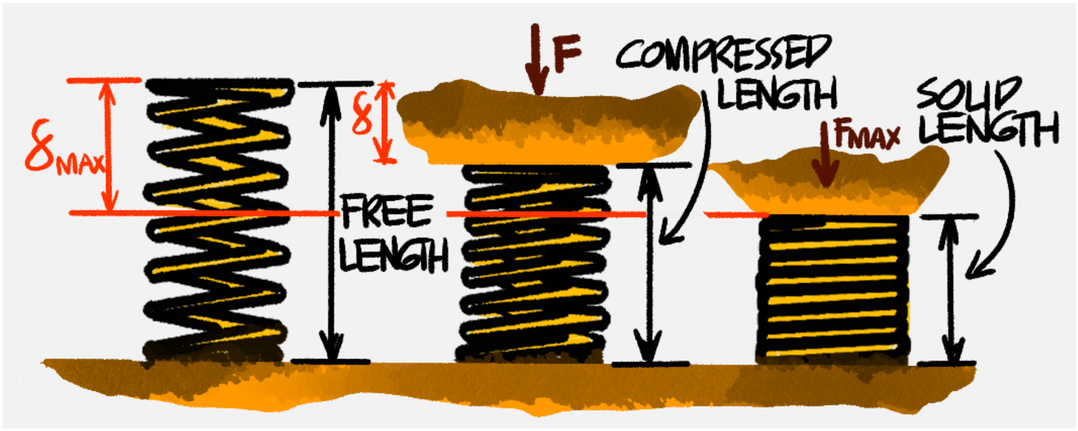
CLOSELY COILED

SPRING IS COILED SO CLOSE THAT THE PLANE CONTAINING EACH TURN IS NEARLY AT RIGHT ANGLES TO THE AXIS OF THE HELIX AND THE **WIRE IS SUBJECTED TO TORSION. HELIX ANGLE IS USUALLY LESS THAN 10 DEGREES**

OPEN COILED

THERE IS A **GAP BETWEEN THE TWO CONSECUTIVE TURNS**, AS A RESULT OF WHICH THE **HELIX ANGLE IS LARGE**

- **INACTIVE COILS** – PART OF THE COIL WHICH IS IN CONTACT WITH THE SEAT AND DOES NOT CONTRIBUTE TO SPRING ACTION.
- **ACTIVE TURNS** – TURNS WHICH IMPART SPRING ACTION.



LENGTH OF WIRE

$$L = \pi D_m n$$

D_m = MEAN DIAMETER
n = EFFECTIVE NO. OF COILS

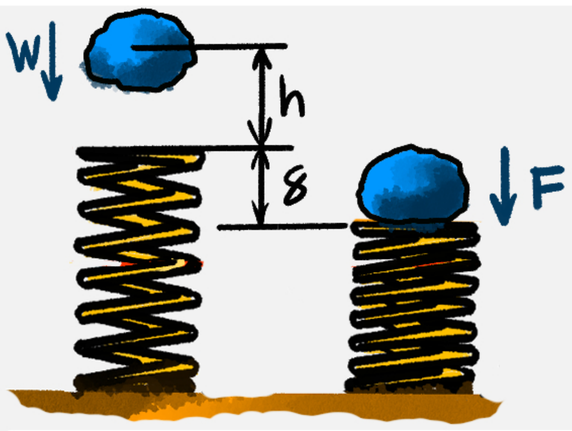
FREE LENGTH

- $FL = SL + \delta_{MAX}$
- $FL = CL + \delta$

FL = FREE LENGTH
SL = SOLID LENGTH
CL = COMPRESSED/ OPERATING LENGTH

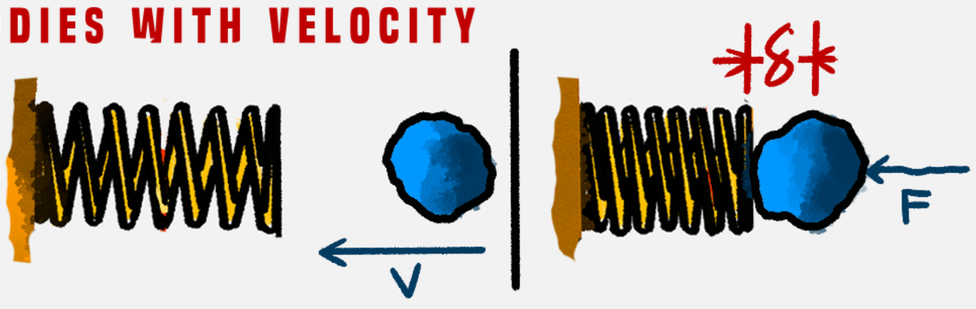
IMPACT LOADING ON SPRINGS

FREE FALL



$$W = (h + \delta) = \delta\left(\frac{F}{2}\right)$$

BODIES WITH VELOCITY



$$\left(\frac{WV^2}{2g}\right) = \left(F\frac{\delta}{2}\right)$$

ENERGY OF A SPRING:
$$E = \frac{1}{2}k\delta^2$$

SPRINGS IN SERIES

$$F_T = F_1 = F_2 = F_3$$
$$\delta_T = \delta_1 + \delta_2 + \delta_3$$
$$k_{TOT} = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3}\right)^{-1}$$

SPRINGS IN PARALLEL

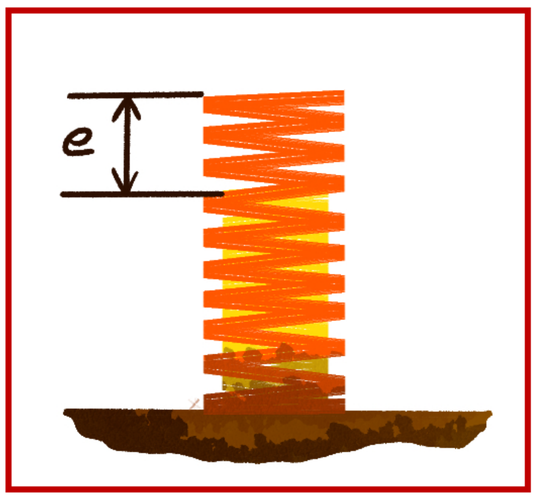
$$F_T = F_1 + F_2 + F_3$$
$$\delta_T = \delta_1 = \delta_2 = \delta_3$$
$$k_{TOT} = k_1 + k_2 + k_3$$

INDETERMINATE SPRINGS

OUTER SPRING LONGER THAN INNER SPRING

- $F = F_o + F_i$
- $\delta_o = e + \delta_i$

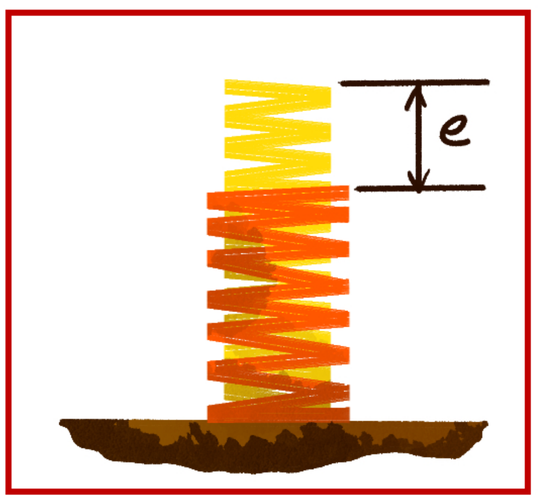
NOTE:
F_o, δ_o = OUTER SPRING
F_i, δ_i = INNER SPRING



INNER SPRING LONGER THAN OUTER SPRING

- $F = F_o + F_i$
- $\delta_i = e + \delta_o$

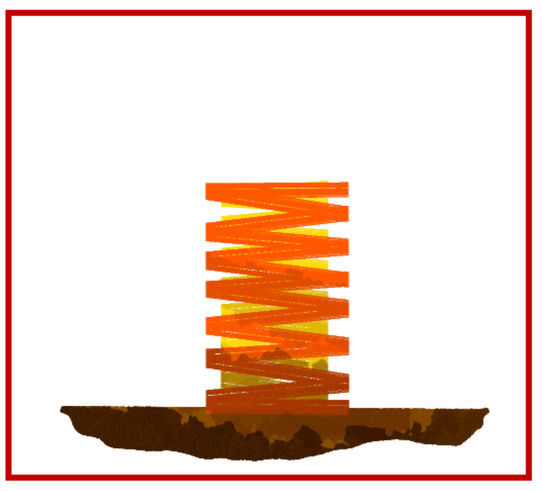
NOTE:
F_o, δ_o = OUTER SPRING
F_i, δ_i = INNER SPRING



SAME HEIGHT

- $F = F_o + F_i$
- $\delta_o = \delta_i$

NOTE:
F_o, δ_o = OUTER SPRING
F_i, δ_i = INNER SPRING



CONICAL / VOLUTE SPRINGS

SPECIAL APPLICATIONS WHERE A TELESCOPING SPRING (SPRING WITH A SPRING RATE THAT INCREASES WITH THE LOAD IS DESIRED).

TORSIONAL SPRINGS

- **HELICAL TYPE** –USED ONLY IN APPLICATIONS WHERE THE LOAD TENDS TO WING UP THE SPRING AND ARE USED IN **VARIOUS ELECTRICAL MECHANISMS.**
- **SPIRAL TYPE** – IS USED WHERE THE LOAD TENDS TO INCREASE THE NUMBER OF COILS AND WHEN MADE OF FLAT STRIP ARE **USED IN WATCHES AND CLOCKS.**

LAMINATED / LEAF SPRINGS

CONSIST OF A NUMBER OF **FLAT PLATES** (KNOWN AS LEAVES) OF VARYING LENGTHS HELD TOGETHER BY MEANS OF CLAMPS AND BOLTS

SIMPLE CANTILEVER (QUARTER ELLIPTICAL LEAF SPRING)

$$6FL = Sbnt^2$$

$$\delta = \frac{SL^2}{Et}$$

IF MULTIPLE:

$$n = (2ng + 3nf)$$

F = LOAD
S = BENDING STRESS
n = NO. OF LEAVES
L = LENGTH
b = WIDTH
t = THICKNESS
ng = NO. OF GRADUATED LEAVES
nf = NO. OF FULL

SEMI-ELLIPTICAL SPRING

$$3FL = S(2bnt^2)$$

$$\delta = \frac{SL^2}{4Et} \quad \text{(W/O EXTRA FULL LENGTH LEAF)}$$

$$\delta = \frac{SL^2}{2(2+r)Et} \quad \text{(W/ EXTRA FULL LENGTH LEAF)}$$

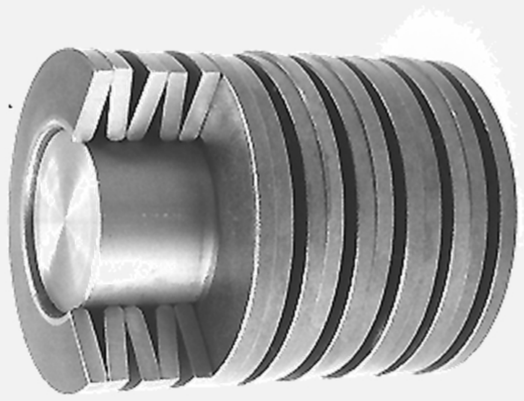
NOTE:

$r = \frac{n'}{n}$ → n' = EXTRA FULL LENGTH LEAVES; n = NO. OF FULL LENGTH LEAVES

F = TOTAL LOAD, $\frac{1}{2}$ OF WHICH IS APPLIED @ EACH END

DISC / BELLEVILLE SPRINGS

CONSIST OF **A NUMBER OF CONICAL DISCS HELD TOGETHER AGAINST SLIPPING BY A CENTRAL BOLT** OR TUBE. THESE SPRINGS ARE USED IN APPLICATIONS WHERE **HIGH SPRING RATES AND COMPACT SPRING UNITS ARE REQUIRED.**



$$\delta = \frac{4F\left[r_o^2 - r_i^2 + r_o r_i \ln\left(\frac{r_o}{r_i}\right)\right]}{3\pi(Et^3)}$$

TORSION SPRINGS

STRESS IN TORSION SPRING

$$S = k_1 \frac{32M}{\pi d^3} = k_1 \frac{32FL}{\pi d^3}$$

k_1 = STRESS FACTOR FOR ROUND WIRE

- **HELICAL TORSION SPRING (ROUND)**

$$\delta = \frac{64FDm a^2 n}{Ed^4}$$

a = MOMENT ARM OF THE LOAD
n = NO. OF EFFECTIVE COILS

- **HELICAL TORSION SPRING (ROUND)**

$$\delta = \frac{64FLa^2}{\pi Ed^4}$$

- **HELICAL TORSION SPRING**

$$\delta = \frac{12FLa^2}{Ebt^3}$$

BELTS

THE BELTS ARE USED TO TRANSMIT POWER FROM ONE SHAFT TO ANOTHER BY MEANS OF PULLEYS WHICH ROTATE AT THE SAME SPEED OR AT DIFFERENT SPEEDS.

TYPES OF BELTS

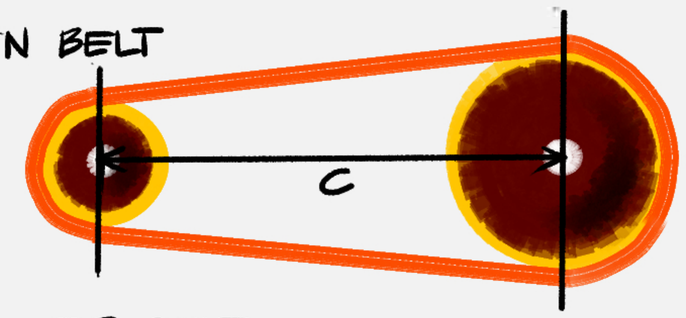
FLAT BELT	MODERATE AMOUNT OF POWER IS TO BE TRANSMITTED, WHEN PULLEYS ARE NOT MORE THAN 8 METRES APART
V BELT	GREAT AMOUNT OF POWER IS TO BE TRANSMITTED, WHEN PULLEYS ARE VERY NEAR TO EACH OTHER.
CIRCULAR BELT OR ROPE	GREAT AMOUNT OF POWER IS TO BE TRANSMITTED, WHEN PULLEYS ARE MORE THAN 8 METRES APART.
TOOTHED BELT	PAIRED WITH TOOTHED PULLEYS AND USED AS TIMING BELT WHERE SPEED RATIO MUST BE MAINTAINED

MATERIALS FOR TRANSMISSION BELTS

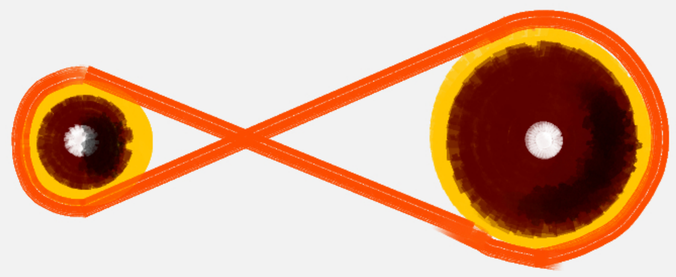
OAK TANNED LEATHER	STANDARD MATERIAL FOR FLAT BELTS
CHROME LEATHER	USED WHERE VERY FLEXIBLE MATERIAL IS DESIRED
RUBBER BELT	USED WHEN EXPOSED TO MOISTURE, ACIDS, AND ALKALIES
FABRIC AND CANVAS BELTS	LIGHT POWER TRANSMISSIONS

FLAT BELTS

- OPEN BELT



- CROSSED BELT



BELT LENGTH

- OPEN BELT CONNECTION

$$L_o = \frac{\pi}{2}(D_1 + D_2) + 2C + \frac{(D_2 - D_1)^2}{4C}$$

- CROSSED BELT CONNECTION

$$L_c = \frac{\pi}{2}(D_1 + D_2) + 2C + \frac{(D_1 + D_2)^2}{4C}$$

FROM CROSSBELT TO OPEN BELT: *LENGTH*

$$L_C - L_o = \frac{D_1 \times D_2}{C}$$

IF C IS NOT GIVEN:

$C = \frac{D_2 + D_1}{2} + D_1$ or $C = D_2$

WHICHEVER IS LARGER !!

ANGLE OF CONTACT

- OPEN BELT CONNECTION

$$\theta = 180^\circ \pm 2\sin^{-1}\left(\frac{D_2 - D_1}{2C}\right)$$

- CROSSED BELT CONNECTION

$$\theta = 180^\circ + 2\sin^{-1}\left(\frac{D_2 + D_1}{2C}\right)$$

NOTE:
+ LARGER ∅
− SMALLER ∅

$\theta = \pi \pm \frac{D_2 - D_1}{C}$

$\theta = \pi + \frac{D_2 + D_1}{C}$

ARC OF CONTACT

$$S = r \times \theta$$

BELT TENSION RATIO

$$\frac{F_1 - F_c}{F_2 - F_c} = e^{f\theta}$$

WHEN F_c IS NOT CONSIDERED

$$\frac{F_1}{F_2} = e^{f\theta}$$

F_c = CENTRIFUGAL FORCE
θ = ANGLE OF CONTACT, rad.
f = COEFF. OF FRICTION

TORQUES

$$T = (F_1 - F_2)r$$

NOTE: R_S = SMALL RADIUS
R_L = LARGE RADIUS

RESISTING TORQUE $= (F_1 - F_2)r_S$

DRIVING TORQUE $= (F_1 - F_2)r_L$

CENTRIFUGAL FORCE

$$F_c = \frac{WV^2}{gr} = \frac{mV^2}{r}$$

$$F_c = \frac{12\rho b t V^2}{g} \quad \text{(ENGLISH)}$$

ρ = DENSITY ; lb/in³
b = WIDTH ; in
t = THICKNESS ; in
V = VELOCITY ; ft/s
F_c = CENTRIFUGAL FORCE ; lbs

EFFECTIVE BELT PULL

$$F_1 - F_2 = (F_1 - F_c)\left[\frac{e^{f\theta} - 1}{e^{f\theta}}\right]$$

f = COEFF. FRICTION
θ = ∠ OF CONTACT
F_1 = TIGHT SIDE
F_2 = SLACK SIDE

POWER TRANSMITTED

$$F_1 - F_2 = bt\left[Sd - \frac{12\rho V^2}{g}\right]\left[\frac{e^{f\theta} - 1}{e^{f\theta}}\right]$$

→ FROM CENTRIFUGAL FORCE

- POWER

$P = (F_1 - F_2)(V)$

- AREA
- DESIGN STRESS

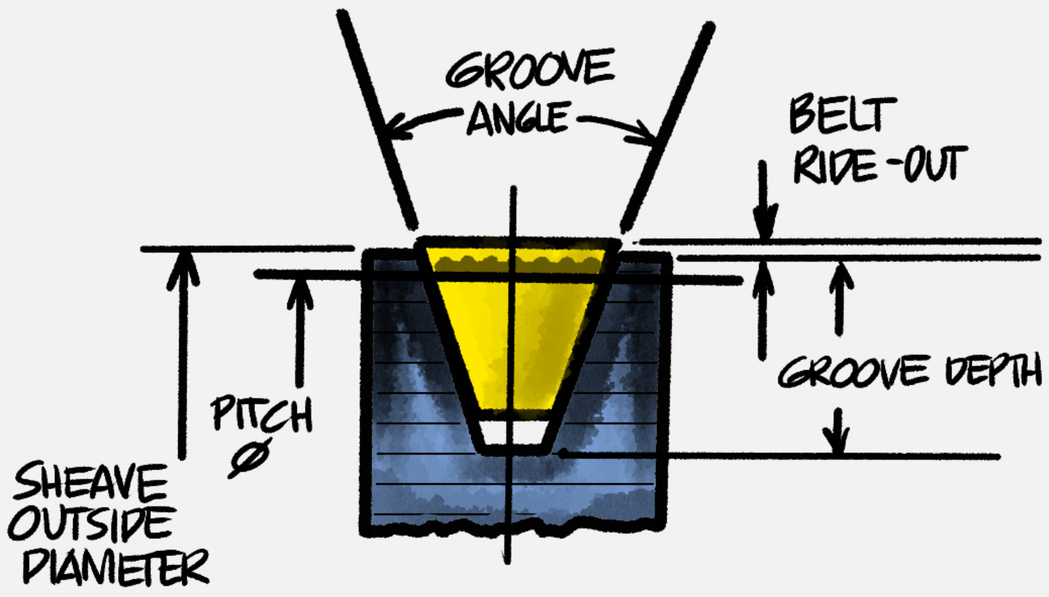
BELT SPEED

$$V = \pi D_1 N_1 = \pi D_2 N_2$$

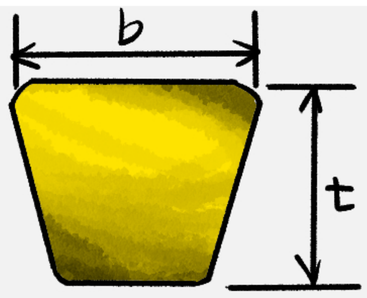
- IF BELT SPEED & PERCENT SLIP IS GIVEN:
  - $V = \pi D_1 N_1 (1 - \%\ \text{SLIP})$ → DRIVER
  - $V = \pi D_2 N_2 (1 + \%\ \text{SLIP})$ → DRIVEN

COEFFICIENT OF FRICTION		DENSITIES	
IRON/STEEL	0.3	LEATHER	0.035 lb/in³
PAPER PULLEY	0.5	RUBBER	0.045 lb/in³

V-BELTS



V-BELTS: STANDARDS



HEAVY DUTY INDUSTRIAL V-BELTS:

EXAMPLE: SIZE A (INCH SIZE)
$b = 1/2''$
$t = 5/16''$

METRIC SIZE = 13C
NOMINAL TOP WIDTH (mm)

LENGTH DESIGNATION

B75 → LENGTH : 76.8″
→ SECTION B V-BELT

HORSEPOWER RATING FOR V-BELTS

$$HP = XS^{0.91} - \frac{YS}{d_e} - ZS^3$$

d_e = EQUIV. ⌀ OF SMALL SHEAVE → TOPITCH ⌀ × SMALL ⌀ FACTOR

X, Y, Z → CONSTANTS
S = BELTSPEED, thousands of feet per minute

$$S = \frac{\pi D N}{1000}$$

STANDARD SIZES OF V-BELTS ; b x t

A-SECTION	1/2″ × 15/16″
B-SECTION	21/32″ × 13/32″
C-SECTION	7/8″ × 17/32″
D-SECTION	1 1/4″ × 3/4″
E-SECTION	1 1/4″ × 29/32″

CLASSIFICATION OF V-BELT DRIVES

V-V DRIVE	BOTH PULLEYS ARE SHEAVES
V-FLAT DRIVE	DRIVING PULLEY IS A SHEAVE WHILE DRIVEN PULLEY HAS A PLANE SURFACE

BELT TENSION RATIOS

$$\left[\frac{F_1 - F_c}{F_2 - F_c}\right] = e^{\frac{f\theta}{\sin\alpha}}$$

θ = ∠ OF CONTACT, rad
$\alpha = \frac{1}{2}$ SHEAVE ∠, deg

EFFECTIVE BELT PULL

$$F_1 - F_2 = (F_1 - F_c)\left[\frac{e^{\frac{f\theta}{\sin\alpha}} - 1}{e^{\frac{f\theta}{\sin\alpha}}}\right]$$

RATED HORSEPOWER FOR STD SIZE A,B,C,D,E

$$\text{RATED HP} = \left[a\left(\frac{10^3}{V_m}\right)^{0.09} - \frac{c}{k_d D_1} - \frac{\varepsilon V_m^2}{10^6}\right]\frac{V_m}{10^3}$$

NOTE: k_d = SMALL DIAMETER FACTORS & a, b, ε → POWER CONSTS.

RATED HORSEPOWER FOR *STD. SIZE B ONLY*

FOR SIZE B V-BELTS
$a = 4.737$
$b = 13.962$
$\varepsilon = 0.0234$

BASE FORMULA:

$$\text{RATED HP} = \left[a\left(\frac{10^3}{V_m}\right)^{0.09} - \frac{c}{k_d D_1} - \frac{\varepsilon V_m^2}{10^6}\right]\frac{V_m}{10^3}$$

ADJUSTED RATED HP

$$\text{ADJUSTED RATED HP} = K_\theta K_L (\text{RATED HP})$$

K_θ = ARC CONTACT FACTOR
K_L = LENGTH CORRECTION FACTOR

DESIGN HP

$$\text{DESIGN HP} = (\text{POWER TRANSMITTED})(N_{sf})$$

NUMBER OF BELTS

$$N_b = \frac{\text{DESIGN HP}}{\text{ADJUSTED HP}}$$

N_b = NO. OF V-BELTS NEEDED

LEATHER BELTS

STRENGTH OF LEATHER

FROM FAIRIES:

$$S_d = 400\eta$$

NOTE:
S_d = in ksi
η = RELATIVE STRENGTH OF JOINT

RELATIVE STRENGTH OF THE JOINT	
CEMENTING	1.0
WIRE LACING	0.88
METAL HOOK	0.35
FROM VALLANCE	0.9

HP CAPACITY OF LEATHER BELTS

$$hp = \left(\frac{hp}{in}\right)(b)(C_m C_p)(C_f, C_{f_2} \ldots)$$

NOTE: $\frac{hp}{in}$ → TABLE 17.1, FAIRIES

C_m – FACTOR FOR DRIVE TYPE
→ ELECTRIC MOTORS = 1
C_p – FACTOR FOR SIZE OF SMALLEST PULLEY
C_f – ENVIRONMENTAL COND.

RUBBER BELTS

$$HP = \frac{V_m b}{2400} \times N_p \times K\theta$$

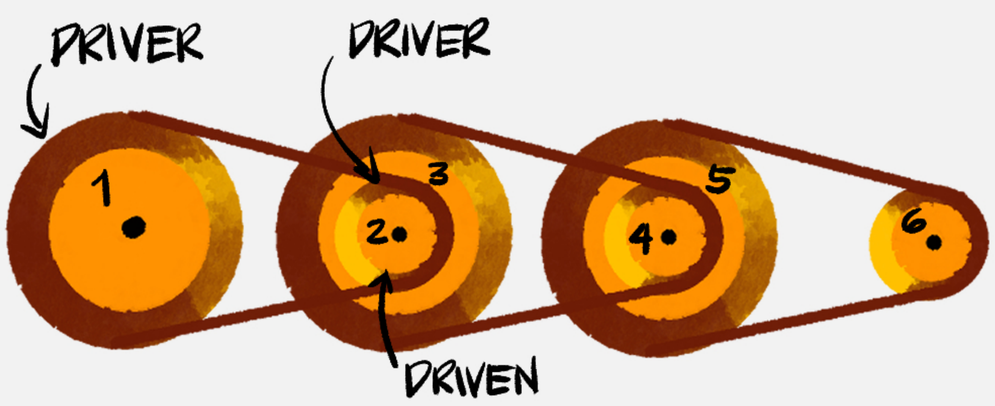
b = BELT WIDTH; in.
N_p = NO. OF PLIES

BELT THICKNESS AND WIDTHS

1/2 in. – 1 in.	INC. BY 1/8
1 in. – 4 in.	INC. BY 1/4
4 in. – 7 in.	INC. BY 1/2
7 in. – 12 in.	INC. BY 1

MAX. BELT WIDTH ($b \leq 8''$)

COMPOUND BELT DRIVES



VELOCITY OF A COMPOUND BELT DRIVE

$$VR = \frac{\text{SPEED OF LAST DRIVEN}}{\text{SPEED OF FIRST DRIVER}} = \frac{\text{PRODUCT OF DRIVER } \varnothing}{\text{PRODUCT OF DRIVEN } \varnothing}$$

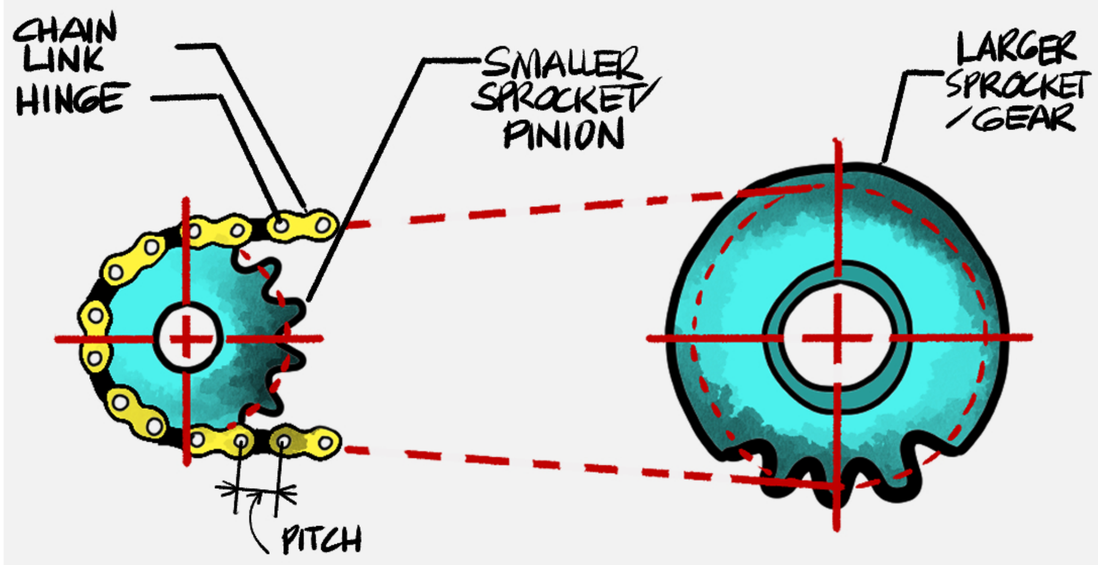
$$VR = \frac{n_6}{n_1} = \frac{D_1 \times D_3 \times D_5}{D_2 \times D_4 \times D_6}$$

STANDARD SIZES OF V-BELTS	
FLAT LEATHER BELTING SPEED	$\leq$ 6000 fpm
V-BELTS SPEED	4500 fpm
LEATHER BELT SPEED	7000 – 8000 fpm
FABRIC BELT SPEED	2000 & MORE fpm
TENSION OF FLAT BELT	3
TENSION OF V-BELT	5

POWER CHAINS AND SPROCKETS

MADE UP OF RIGID LINKS, HINGED TOGETHER BY PIN JOINTS TO PROVIDE THE FLEXIBILITY FOR WRAPPING ROUND THE WHEELS (SPROCKET WHEELS OR SPROCKETS). THE SPROCKETS AND THE CHAIN MOVE TOGETHER WITHOUT SLIPPING AND ENSURES **PERFECT VELOCITY RATIO**. USED FOR TRANSMISSION OF POWER, WHEN CENTER DISTANCE OF SHAFTS IS SHORT.

SPROCKETS AND CHAIN



TYPES OF POWER TRANSMISSION CHAINS

BLOCK OR BUSH CHAIN	USED IN THE EARLY STAGES OF DEVELOPMENT IN THE POWER TRANSMISSION
BUSH ROLLER CHAIN	A PIN PASSES THROUGH THE BUSH WHICH IS SECURED IN THE HOLES OF THE ROLLER. THE ROLLERS ARE FREE TO ROTATE ON THE BUSH WHICH PROTECTS THE SPROCKET AGAINST WEAR. MADE OUT OF ALLOY STEEL.
SILENT CHAIN (INVERTED TOOTH CHAIN)	DESIGNED TO ELIMINATE THE BAD EFFECTS CAUSED BY STRETCHING AND TO PRODUCE NOISELESS RUNNING. IT CORRECTS THE SMALL CHANGE IN THE PITCH

TYPES OF SPROCKETS

TYPE A	PLAIN PLATE	**TYPE C**	HUB ON BOTH
TYPE B	HAS HUB ON ONE SIDE	**TYPE D**	DETACHABLE HUB

CHAIN NO.	25	35	40	41	50	60	80	100	120	140	160	180	200	240
PITCH, IN	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	3

PITCH DIAMETER OF SPROCKET

$$D = \frac{P}{\sin\left(\frac{180}{N}\right)}$$

P = PITCH
N = NUMBER TEETH OF SPROCKET

OUTSIDE DIAMETER OF SPROCKET

$$D_o = P\left[0.60 + \cot\left(\frac{180}{N}\right)\right]$$

LENGTH OF CHAIN

$$L = 2C + \frac{N_1 + N_2}{2} + \frac{(N_2 - N_1)^2}{40C}$$

C = CENTER DISTANCE

CENTER DISTANCE BETWEEN SPROCKETS

$$C = \frac{P}{8}\left[(2L - N_2 - N_1) + \sqrt{(2L - N_2 - N_1)^2 - 0.810(N_2 - N_1)^2}\right]$$

MAXIMUM PITCH

$$P \leq \left(\frac{900}{n}\right)^{\frac{2}{3}}$$

P = PITCH IN INCHES
n = SPEED OF SMALL SPROCKET

TYPES OF LUBRICATION

TYPE I	MANUAL LUBRICATION	**TYPE III**	BATH OR DISC LUBRICATION
TYPE II	DRIP LUBRICATION	**TYPE IV**	OIL STREAM LUBRICATION

TYPES OF LUBRICATION	
TEMPERATURE	**VISCOSITY**
20 - 40°F	SAE 20
40 - 100°F	SAE 30
100 - 120°F	SAE 40
120 - 140°F	SAE 50

NO. OF TEETH	
LOW SPEEDS	**12**
MODERATE SPEEDS	**17**
HIGH SPEEDS	**21**
MAXIMUM QUIETNESS	**27**

GENERAL RECO FOR SPROCKET DESIGN

- MINIMUM NUMBER OF TEETH SHOULD BE **17** UNLESS THE DRIVE IS AT A VERY LOW SPEED (**100 RPM**).
- MAXIMUM SPEED RATIO SHOULD BE **7**
- CENTER DISTANCE SHOULD BE **30 TO 30 TIMES** THE PITCH
- SMALLER SPROKCET: ARC OF CONTACT SHOULD **NOT BE LESSER THAN 120 DEGREES**
- LARGER SPROKCET: **NO MORE THAN 120 TEETH**

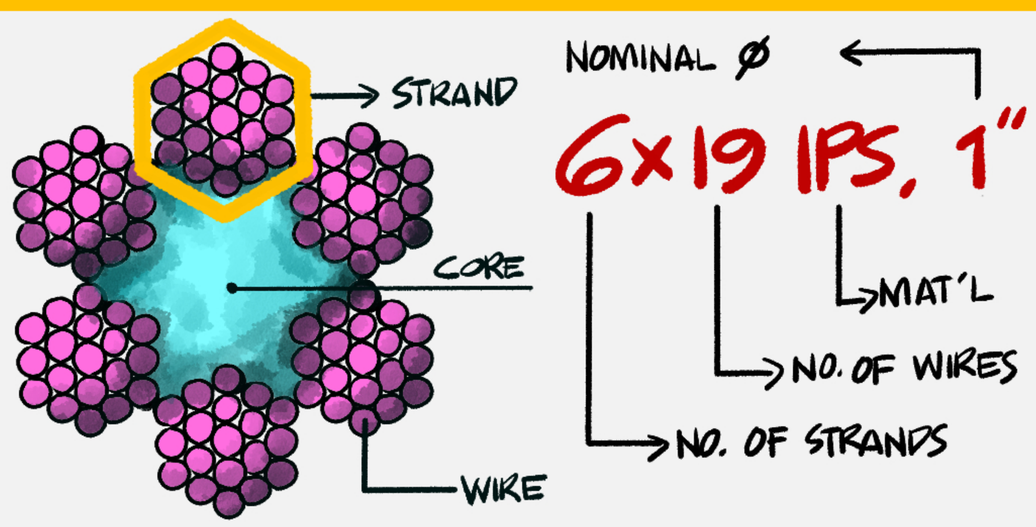
WIRE ROPES

COLD-DRAWN WIRES THAT ARE **FIRST WRAPPED INTO STRANDS**; THE STRANDS ARE THEN WRAPPED INTO HELICES ABOUT A CORE OR CENTRAL ELEMENTS, WHICH IS USUALLY **HEMP OR PULP**.

REGULAR LAY	TWISTED IN **OPPOSITE** DIRECTIONS
LANG LAY	TWISTED IN **SAME** DIRECTIONS
NON - PERFORMED	WIRES AND STRANDS ARE BENT INTO PLACE, RESULTING IN **HIGH STRESSES** IN STRAIGHT, **UNLOADED ROPE**
PERFORMED	INDIVIDUAL STRANDS ARE **MECHANICALLY SHAPED AHEAD OF TIME** INTO THE HELICAL CONFIGURATION THEY HAVE IN THE ROPE

6X7	HAULAGE, TRAMWAYS, GUY WIRES
6X19	GENERAL PURPOSE ROPE, HOISTS, CRANES
6X37	HIGH SPEED ELEVATORS, CRANES, HOISTS
8X19	EXTRA FLEXIBLE HOISTING ROPE APPLICATIONS

WIRE ROPE NOMENCLATURE



DIRECT TENSILE STRESS

$$S_t = F_t / A_r$$

F_t → TOT. TENSION of ROPE
A_r → CROSS SEC. AREA ROPE

TOTAL TENSILE LOAD

$$F_t = \left(1 + \frac{a}{g}\right)(W_L + W_r) + F_b$$

W_L = WT. OF LOAD
W_r = WT. OF ROPE
a = ACCELERATION

WEIGHT OF ROPE

$$W = wL$$

L = LENGTH OF ROPE, ft
w = UNIT WT. OF ROPE ; lb./ft

EMPIRICAL FORMULA: (6x19 WIRE ROPES)

UNIT WEIGHT OF ROPE

FROM COLLINS:

$$w = 1.6 d_r^2 \quad \text{or} \quad w = 1.58 d_r^2$$

APPX. METAL CROSS SECTION

$$A_r \approx 0.4 d_r^2$$

FROM MACHINERY'S : (6x19)
- FIBER CORE ROPE $= 0.4 d_r^2$
- IWRC $= 0.475 d_r^2$

APPX DIAMETER OF OUTER WIRE

FROM COLLINS:

$$d_w = \frac{d_r}{13} \text{ to } \frac{d_r}{16}$$

FROM MACHINERY'S : (6x19)

$$d_w = 0.063 d_r$$

RECOMMENDED MINIMUM SHEAVE

FROM COLLINS:

$$d_s = 34 d_r$$

BENDING LOAD

$$S_b = \frac{F_b}{A_r} = E_r\left(\frac{d_w}{d_s}\right)$$

d_w = WIRE ø
d_s = SHEAVE ø
$E_r \approx 12 \times 10^6$ psi (STEEL)

ULTIMATE STRENGTH

$$F_u = 76{,}000\, d_r^2$$

NOTE: FU IS THE SAME FOR 6X7, 6X19, 6X37

FACTOR OF SAFETY

$$N_{sf} = \frac{F_u - F_b}{F_t}$$

COMMON N_{sf}: SMALL HOIST - 7
GUYS - 3.5 HOT LADDLE CRANE - 8
MISC. HOISTING EQUIP. - 5
HAULAGES, CRANES, DERRICKS - 6

ELEVATORS

DETERMINING NUMBER OF ELEVATORS

FROM PSME CODE:

- ☑ NO. OF ELEVATORS $= \frac{\text{NO. OF PERSONS}}{220}$
- ☑ NO. OF PERSONS $= \frac{A_F}{9.3}$

NOTE: A_F → FLOOR AREA ; m^2

MAXIMUM RATED LOAD

PLATFORM AREA $\leq 4.65\ m^2$

$$W = 35.10(A^2) + 326.224(A) \text{ ; kg}$$

MAXIMUM RATED LOAD

PLATFORM AREA $> 4.65\ m^2$

$$W = 2.458(A^2) + 611.36(A) - 621.4 \text{ ; kg}$$

BREAKING STRENGTH (RULE OF THUMB)

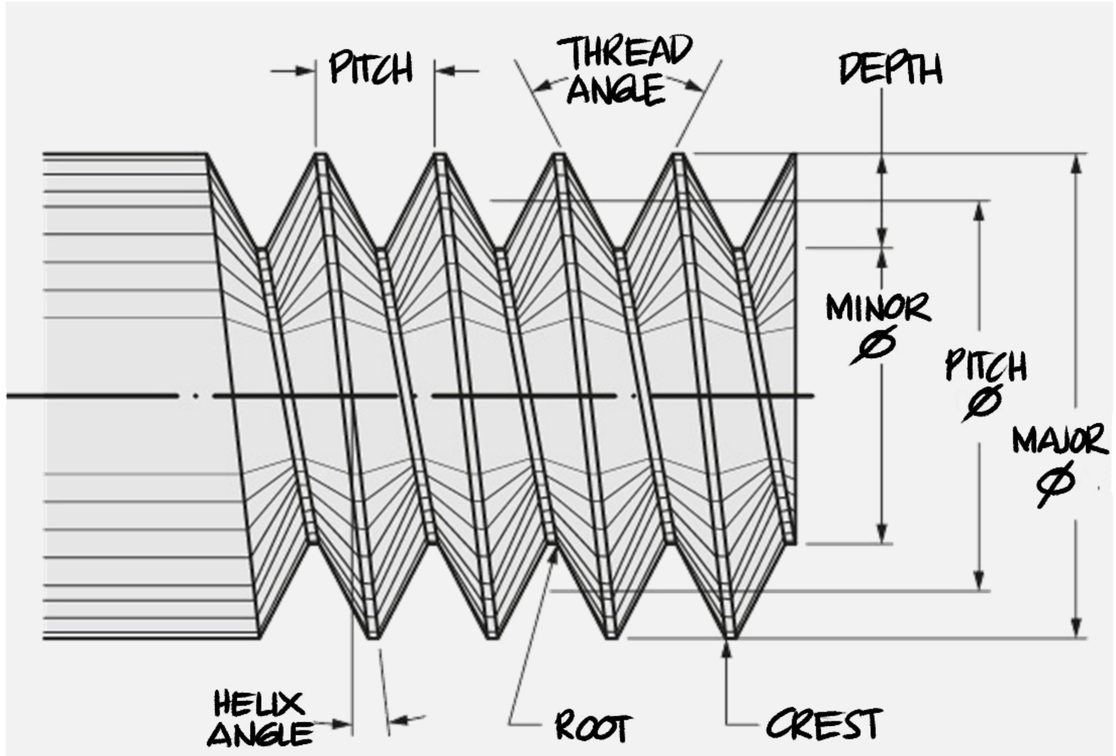
$$B.S. = \frac{D^2}{20}$$

B.S. → BREAKING STRENGTH ; TONS
D → ø IN mm.

SCREW FASTENINGS

SCREW FASTENING	COMPOSED BY A BOLT AND NUT.
SCREW THREAD	FORMED BY CUTTING A CONTINUOUS HELICAL GROOVE ON A CYLINDRICAL SURFACE
BOLT	DESIGNED TO PASS THROUGH HOLES IN THE MATING MEMBERS AND SECURED BY TIGHTENING A NUT (*NUT NEEDED ;(*)
SCREW	INSERTED THROUGH A HOLE IN ONE MEMBER TO BE JOINED AND INTO A THREADED HOLE IN THE MATING MEMBER (*NO NUT NEEDED ;)*)

THE SCREW THREAD



MAJOR DIAMETER (OUTSIDE/ NOMINAL DIAMETER)	LARGEST DIAMETER OF AN EXTERNAL OR INTERNAL SCREW THREAD. *THE SCREW IS SPECIFIED BY THIS DIAMETER*
MINOR DIAMETER (CORE/ROOT DIA.)	SMALLEST DIAMETER OF AN EXTERNAL OR INTERNAL SCREW THREAD.
PITCH DIAMETER (EFFECTIVE DIA.)	DIAMETER OF AN IMAGINARY CYLINDER. IT IS THE MEAN DIAMETER OF MAJOR AND MINOR DIAMETERS
PITCH	DISTANCE FROM A POINT ON ONE THREAD TO THE NEXT. MEASURED IN AN AXIAL DIRECTION BETWEEN CORRESPONDING POINTS.
LEAD	DISTANCE BETWEEN TWO POINTS ON THE SAME HELIX. THE DISTANCE WHICH A SCREW THREAD ADVANCES AXIALLY IN ONE ROTATION OF THE NUT.
CREST	IS THE TOP SURFACE OF THE THREAD
ROOT	BOTTOM SURFACE CREATED BY THE TWO FLANKS
DEPTH OF THREAD	IS THE PERPENDICULAR DISTANCE BETWEEN THE CREST AND ROOT
FLANK	SURFACE JOINING THE CREST AND ROOT
ANGLE OF THREAD	ANGLE INCLUDED BY THE FLANKS OF THE THREAD
SLOPE	HALF THE PITCH OF THE THREAD

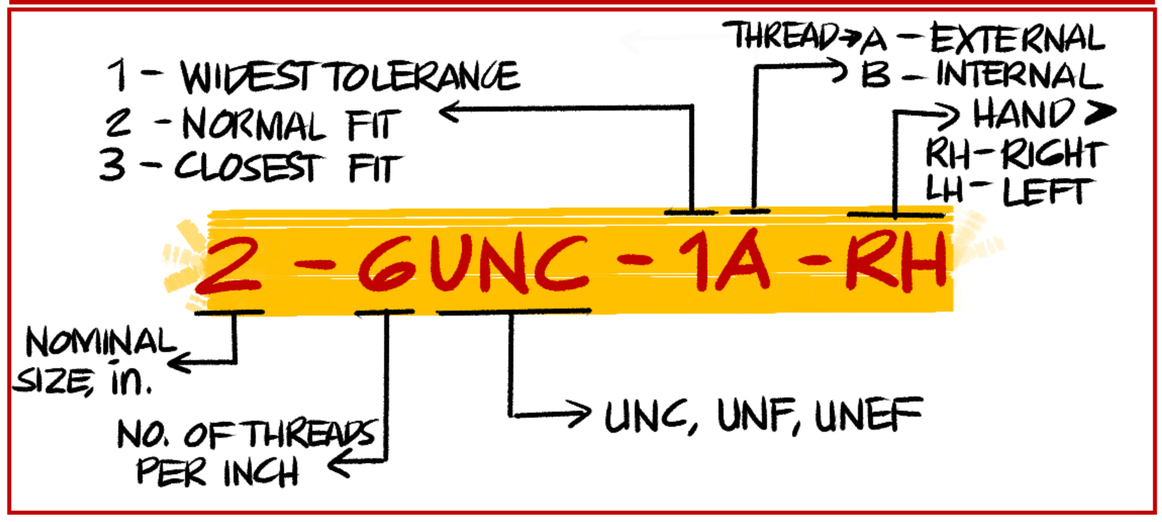
TYPES OF THREADS

U N C **(UNIFIED NATIONAL COARSE SERIES)**	RECOMMENDED FOR GENERAL USE
U N F **(UNIFIED NATIONAL FINE SERIES)**	AUTOMOBILE AND AIRCRAFT WORK
U N E F **(UNIFIED NATIONAL EXTRA FINE)**	AERONAUTICAL EQUIPMENT
8 U N **(UNIFIED 8 – THREAD SERIES)**	BOLTS FOR HIGH-PRESSURE PIPE FLANGES, CYLINDER HEAD, STUDS, ETC.

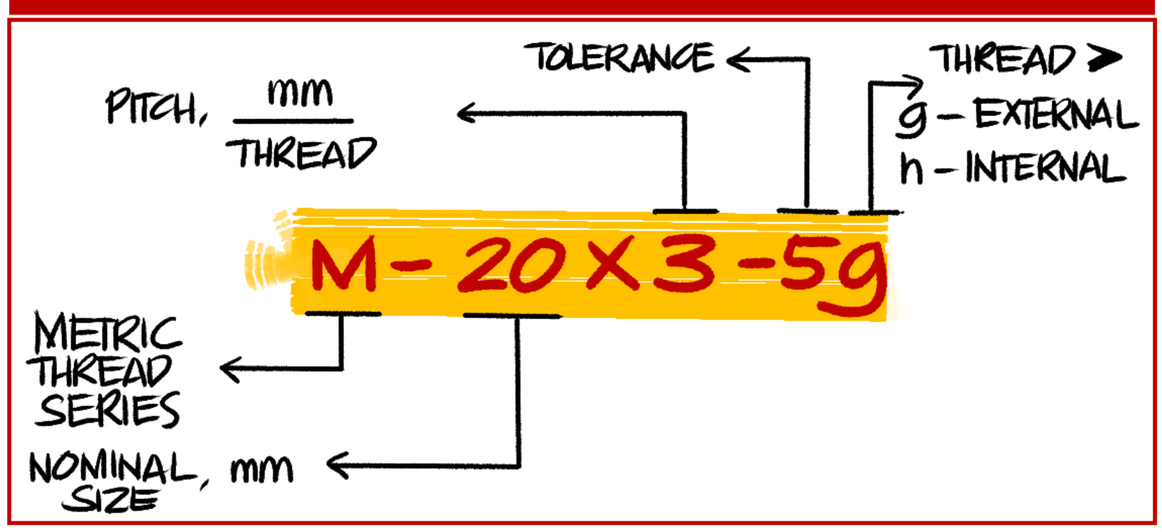
TOLERANCES

• ***CLASS 1A*** (EXTERNAL) • ***CLASS 1B*** (INTERNAL)	WIDEST TOLERANCES WHERE QUICK AND EASY ASSEMBLY, EVEN BRUISED THREADS
• ***CLASS 2A*** (EXTERNAL) • ***CLASS 2B*** (INTERNAL)	TOLERANCES FOR NORMAL PRODUCTION, MOST WIDELY USED
• ***CLASS 3A*** (EXTERNAL) • ***CLASS 3B*** (INTERNAL)	CLOSEST FIT, ALLOWANCE IS ZERO

AMERICAN AND UNIFIED NATIONAL THREAD SERIES

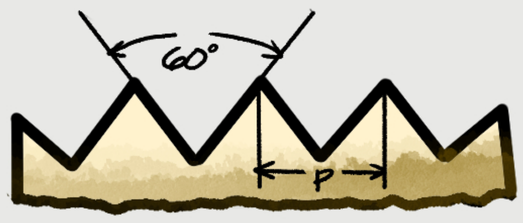


METRIC THREAD SERIES

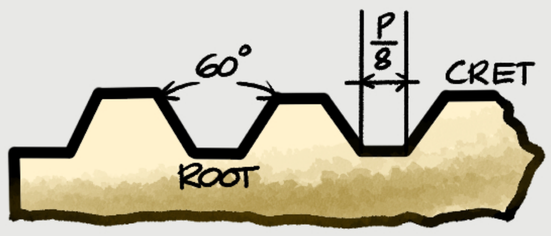


FORM OF THREADS FOR BOLTS

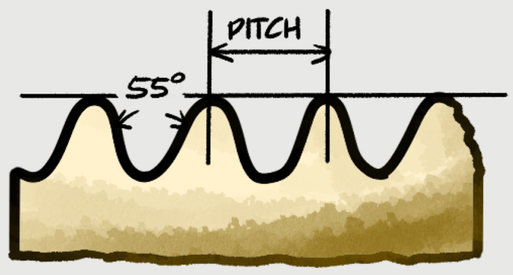
V-THREADS



SELLER SCREW THREAD



WITHWORTH THREAD



SHEAR STRESS (TORSIONAL IN THE ROOT)

$$S_s = \frac{16T}{\pi D_i^3}$$ | D_i = INSIDE Ø

TENSILE STRESS

$$S_t = \frac{F}{A_r} = \frac{F}{A_s}$$ | A_r = ROOT AREA; A_s = STRESS AREA

COMPRESSION STRESS

$$S_c = \frac{F}{\frac{\pi}{4}(D_o^2 - D_i^2)N_T}$$ | D_o – OUTSIDE Ø; D_i – INSIDE Ø; N_T – NO. OF THREADS

SHEAR STRESS

$$S_s = \frac{F}{\pi D_i t N_t}$$ | t = THICKNESS

WORKING STRENGTH OF THE BOLT

$$W = S_t[0.55d^2 - 0.25d]$$

W = WORKING STRENGTH; lb
S_t = ALLOW. WORKING STRESS; psi
d = BOLT Ø; in.

BOLT SPACING

$$Z = \frac{\pi D_{bc}}{N_T}$$ | D_{bc} – BOLT CIRCLE Ø; N_T – TOTAL NO. OF BOLTS; Z – BOLT SPACING

BOLT CIRCLE DIAMETER

$$D_{bc} = D_i + 2D_b$$ | D_i = INSIDE Ø OF CYLINDER; D_b = BOLT Ø

INITIAL TENSION

$$F_i = 16{,}000D$$ | D = NOMINAL Ø; mm or in

BOLT ELONGATION

$$\delta_b = (\text{LEAD})(\text{NO. OF TURNS})$$

POWER AND TORQUE TRANSMITTED BY A SINGLE SET SCREW

$$\text{POWER} = \frac{DNd^{2.3}}{50}$$ | $T = 1250Dd^{2.3}$ | D = SHAFT Ø; in; d = SCREW Ø; in; N = SPEED; rpm; P = hp; T = lb·in

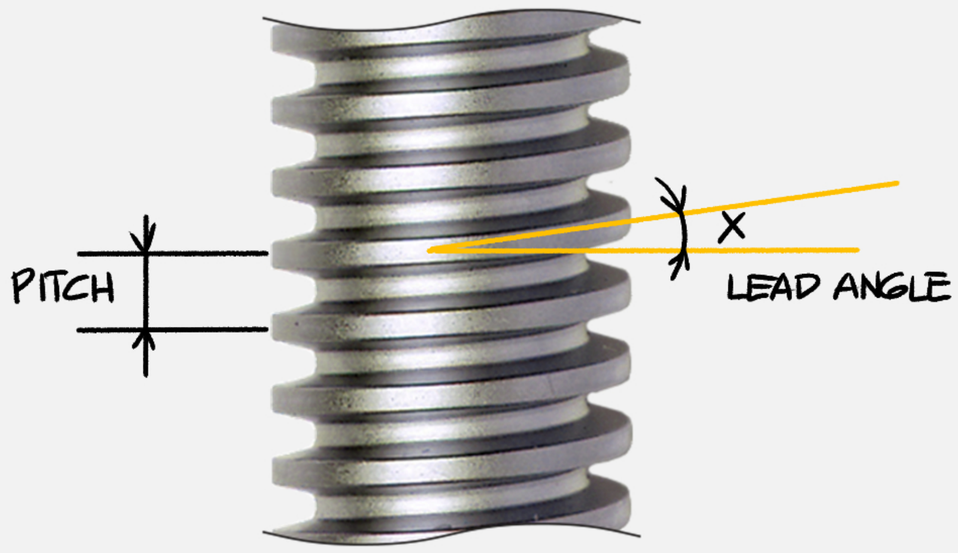
FORMULAS FROM VALLANCE AND FAIRES

	VALLANCE	FAIRES
TENSILE STRESS IN BOLTS	$S_W = C(A_r)^{0.418}$; psi $F_a = C(A_r)^{1.418}$; lbs C = 5,000 CARBON STEEL C = 10,000 BRONZE C = 15,000 ALLOY STEEL A_r = STRESS AREA; in^2	$S_d = \frac{S_y}{6}(A_s)^{0.5}$ $F_e = \frac{S_y}{6}(A_s)^{1.5}$ S_d = DESIGN TENSILE STRESS; psi F_e = TENSILE LOAD S_y = YIELD STRENGTH IN TENSION, PSI
DEPTH OF TRAP	☑ CAST IRON : 1.5D ☑ CAST STEEL : 1.25D D = NOMINAL Ø	☑ CAST IRON: 1.5D ☑ STEEL & WROUGHT IRON: D D = NOMINAL Ø
INITIAL TENSION AND TORQUE	$T = 0.2F_aD$ WHERE: T = TORQUE ; lb-in or N-mm F_a = APPLIED LOAD; lb or N D = NOM. Ø; in or mm	$T = CDF_i$ C → 0.2 RECEIVED 0.15 LUBRICATED F_i = INITIAL TENSION; lb or N

POWER SCREWS

MOVE MACHINE PARTS AGAINST RESISTING FORCES, FOR INSTANCE, IN A SCREW-OPERATED TENSILE-TESTING MACHINE, JACK, PRESS, OR LEAD SCREW OF A LATHE.

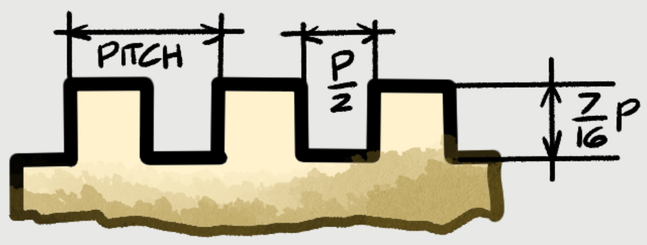
POWER SCREW



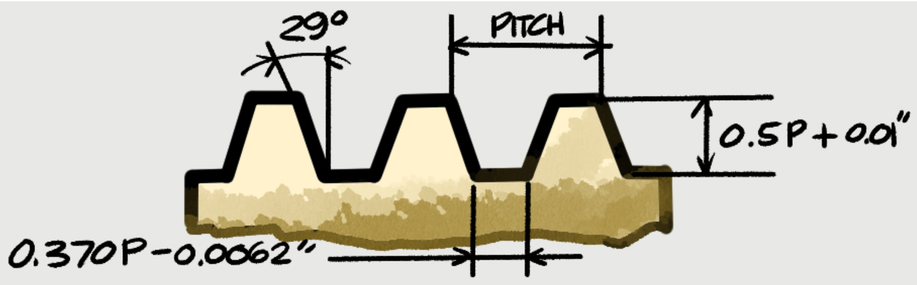
CONVENTIONAL POWER SCREWS: 30-70 %
BALL SCREWS: 90 %

FORM OF THREADS FOR POWER SCREWS

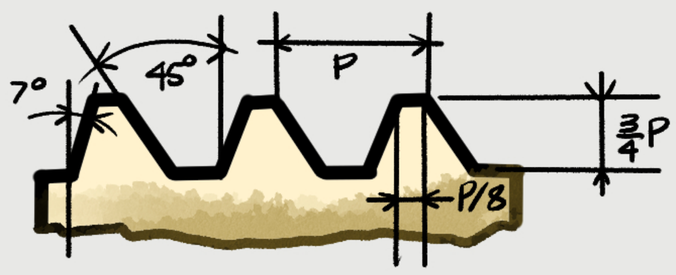
SELLERS SQUARE



ACME THREAD



BUTTRESS THREAD



EFFICIENCY

$$e = \frac{P_o}{P_i} = \frac{WV}{2\pi TN} = \frac{\tan\alpha}{\tan(\alpha+\theta_f)+\frac{f_c D_c}{D_m}}$$

→ D_c = COLLAR Ø
D_m = MEAN Ø SCREW

LINEAR VELOCITY

$$V = N \cdot L$$

N → ROTATIVE SPEED
L → LEAD

LEAD ANGLE

$$\alpha = \tan^{-1}\left(\frac{L}{\pi D_m}\right)$$

➢ LEAD
SINGLE THREAD L = P
DOUBLE THREAD L = 2P
TRIPPLE THREAD L = 3P

OUTSIDE DIAMETER

$$D_o = D_i + \frac{L}{2}$$

D_o = OUTSIDE Ø
D_i = INSIDE Ø
L = LEAD

TORQUE REQUIRED TO OVERCOME COLLAR FRICTION

$$T_c = \frac{f_c W (D_c)}{2} = \frac{f_c W (r_o + r_i)}{2}$$

W = LOAD
f = COEFF. FRICTION
D_c = COLLAR Ø
$D_c = \frac{D_o + D_i}{2}$

TORQUE REQUIRED TO OVERCOME THREAD FRICTION

RAISING THE LOAD

$$T_f = \frac{WD_m}{2}\left[\frac{b\tan\alpha + f}{b - f\tan\alpha}\right]$$

LOWERING THE LOAD

$$T_f = \frac{WD_m}{2}\left[\frac{b\tan\alpha - f}{b + f\tan\alpha}\right]$$

GENERAL:

$$P_i = 2\pi T_{TOT} N$$

$$T_{TOT} = T_c + T_f$$

T_f → THREAD FRICTION
T_c → OVERCOME COLLAR FRICTION

REMEMBER ME!!!

b = 1	SQUARE	cos(0°)
b = 0.968	ACME	cos(14.5°)
b = 0.966	TRAPEZOIDAL METRIC	cos(15°)
b = 0.866	AMERICAN STD. COURSE	cos(30°)

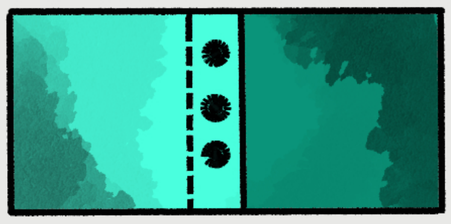
RIVETED JOINTS

SHORT CYLINDRICAL BAR WITH A HEAD INTEGRAL TO IT. THE RIVETS ARE USED TO MAKE PERMANENT FASTENING BETWEEN THE PLATES SUCH AS IN STRUCTURAL WORK, SHIP BUILDING, BRIDGES, TANKS AND BOILER SHELLS. THE RIVETED JOINTS ARE WIDELY **USED FOR JOINING LIGHT METALS.**

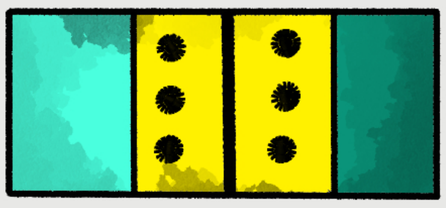
TYPES OF RIVETED JOINTS

DEPENDING UPON THE WAY IN WHICH THE *PLATES ARE CONNECTED*

LAP JOINT



BUTT JOINT



- SINGLE STRAP BUTT JOINT

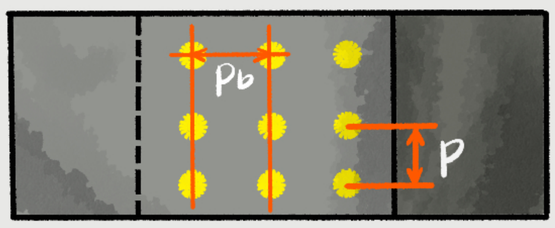
- DOUBLE STRAP BUTT JOINT

TYPES OF RIVETED , JOINTS DEPENDING UPON THE *NUMBER OF ROWS OF THE RIVETS*

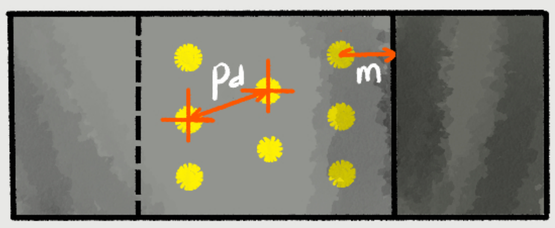
SINGLE RIVETED

DOUBLE RIVETED

CHAIN RIVETED



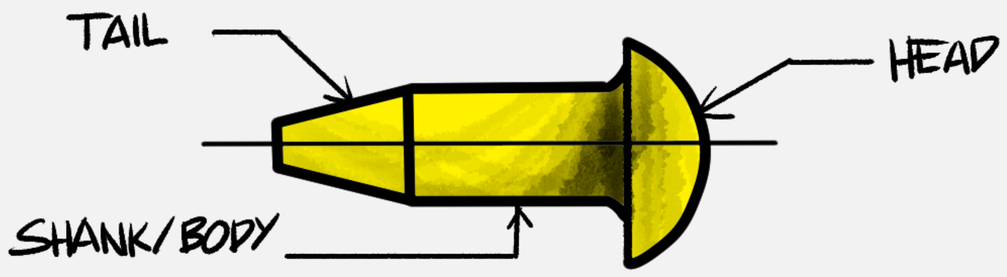
ZIG-ZAG RIVETED



SHARP V-THREAD

$$\text{DEPTH OF THREAD} = p \times \cos 30^\circ = \frac{\cos 30^\circ}{\text{NO. OF THREAD PER INCH}}$$

RIVET



PITCH, p — FROM THE CENTER OF ONE RIVET TO THE NEXT RIVET MEASURED PARALLEL TO THE SEAM

BACK PITCH, P_b — PERPENDICULAR DISTANCE BETWEEN THE CENTER LINES OF THE SUCCESSIVE ROWS

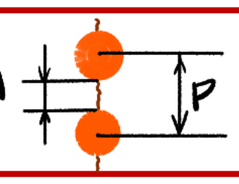
DIAGONAL PITCH, P_d — BETWEEN THE CENTERS OF THE RIVETS IN ADJACENT ROWS OF **ZIG-ZAG RIVETED JOINT**

MARGIN OR MARGINAL PITCH, m — DISTANCE BETWEEN THE CENTER OF RIVET HOLE TO THE NEAREST EDGE OF THE PLATE

TEARING OF THE PLATE ACROSS A ROW OF RIVETS.

$$F_t = S_t \times A_t = S_t \times (p-d)(t)$$

S_t = PERMISSIBLE TENSILE STRESS



SHEARING OF RIVETS

$$F_s = (S_s)(\pi/4 \times d^2 \times t \times n) \longrightarrow \text{SINGLE SHEAR}$$
$$F_s = (S_s)(\pi/4 \times d^2 \times t \times n \times 2) \longrightarrow \text{DOUBLE SHEAR}$$

n - NO. OF BOLTS

CRUSHING OF THE PLATE OR RIVETS

$$F_c = S_c \times (d \times t \times n) \longrightarrow \text{SAME AS BEARING}$$

EFFICIENCY OF A RIVETED JOINT

$$e = \frac{\text{LOAD THAT WILL PRODUCE ALLOWABLE STRESS IN JOINTS}}{\text{LOAD THAT WILL PRODUCE THE ALLOWABLE TENSION STRESS IN THE UNPUNCHED PLATE}} = \frac{\text{LEAST of } F_t, F_s, F_c}{(S_{tm})(p \times t)}$$

S_{tm} → TENSILE STRENGTH OF MATERIAL

JOINT STRENGTH

FROM MACHINERY'S:

$$F = n \times A_s \times S_s$$

n - NO. OF RIVETS
A_r - CROSS SEC. AREA OF RIVETS
S_s - ALLOW. SHEARING STRESS

RIVET DIAMETER

$$d = 1.2\sqrt{t} \text{ to } 1.4\sqrt{t}$$

TENSILE STRESS AREA : STEELS $S_{ULT.} < 100{,}000$psi

$$A_t = \frac{\pi}{4}\left(d_{maj} - \frac{0.9743}{n}\right)^2 = \frac{\pi}{4}\left(D - \frac{\cos 13}{n}\right)^2$$

TENSILE STRESS AREA : STEELS $S_{ULT.} > 100{,}000$psi

$$A_t = \pi\left(\frac{d_{min}}{2} - \frac{0.16238}{n}\right)^2 = \pi\left(\frac{d_{min}}{2} - \frac{\cos 13}{6n}\right)^2$$

d_{maj} → BASIC MAJ. ø d_{min} → MIN. PITCH ø OF EXT. THREAD

WELDED JOINTS

PERMANENT JOINT OBTAINED BY THE FUSION OF THE EDGES OF THE TWO PARTS TO BE JOINED TOGETHER, WITH OR WITHOUT THE APPLICATION OF PRESSURE AND A FILLER MATERIAL.

WELDING	HEATING THE METAL TO A STATE OF FUSION PERMITTING IT TO FLOW TOGETHER INTO A SOLID JOINT
1. FUSION WELDING	PARTS TO BE JOINTED ARE HELD IN POSITION WHILE THE MOLTEN METAL IS SUPPLIED TO THE JOINT. MOLTEN METAL MAY COME FROM THE PART ITSELF (**PARENT METAL**) OR FILLER METALS.
• **THERMIT WELDING**	MIXTURE OF **IRON OXIDE AND ALUMINIUM** CALLED *THERMIT* IS IGNITED AND THE IRON OXIDE IS REDUCED TO MOLTEN IRON.
• **GAS WELDING**	APPLYING THE FLAME OF AN **OXY-ACETYLENE OR HYDROGEN GAS** FROM A WELDING TORCH. THE OPERATOR **USES A WELDING ROD TO SUPPLY THE METAL FOR THE WELD**. *FLUX* IS USED TO REMOVE THE SLAG.
• **ELECTRIC ARC WELDING**	HEAT SUPPLIED BY A **CONTINUOUS ARC DRAWN BETWEEN TWO ELECTRODES**, THE WORK FORMS ONE ELECTRODE AND THE WELDING ROD FORMS THE OTHER.
- UN-SHIELDED ARC WELDING	WHEN A LARGE ELECTRODE OR FILLER ROD IS USED
- SHIELDED ARC WELDING	WELDING RODS COATED WITH SOLID MATERIAL
2. FORGE WELDING	THE PARTS TO BE JOINTED ARE **FIRST HEATED TO A PROPER TEMPERATURE IN A FURNACE OR FORGE AND THEN HAMMERED**. THIS METHOD OF WELDING IS RARELY USED NOW-A-DAYS. AN ELECTRIC-RESISTANCE WELDING IS AN EXAMPLE OF FORGE WELDING.
3. SOLDERING	USING AN ALLOY OF **LEAD AND TIN (CALLED SOLDER)** APPLIED BETWEEN THE TWO PIECES IN A MOLTEN STATE
4. BRAZING	USING A NON-FERROUS FILLER **(COPPER ALLOY)** WHICH IS MELTED AND APPLIED TO THE PIECES BEING JOINED

CODE NUMBER OF STEEL ELECTRODES

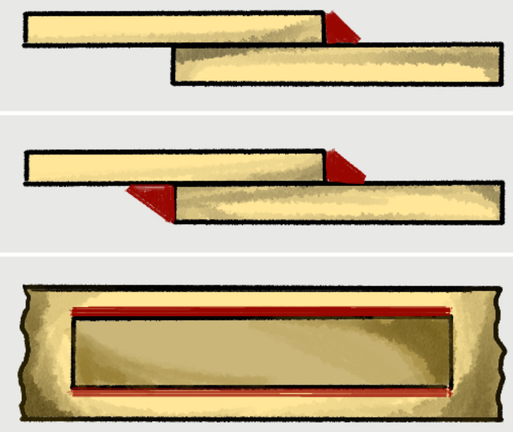
➢ BY ASTM & AWS

LAP JOINT / FILLET JOINT - OVERLAPPING THE PLATES AND THEN WELDING THE EDGES OF THE PLATES. THE CROSS-SECTION OF THE FILLET IS APPROXIMATELY TRIANGULAR

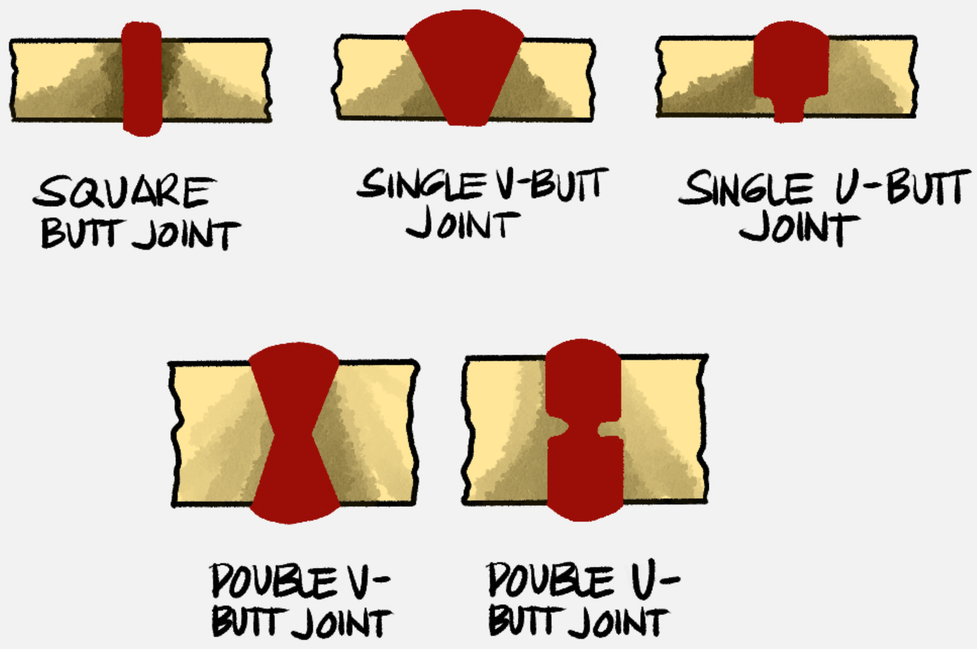
SINGLE TRANSVERSE

DOUBLE TRANSVERSE

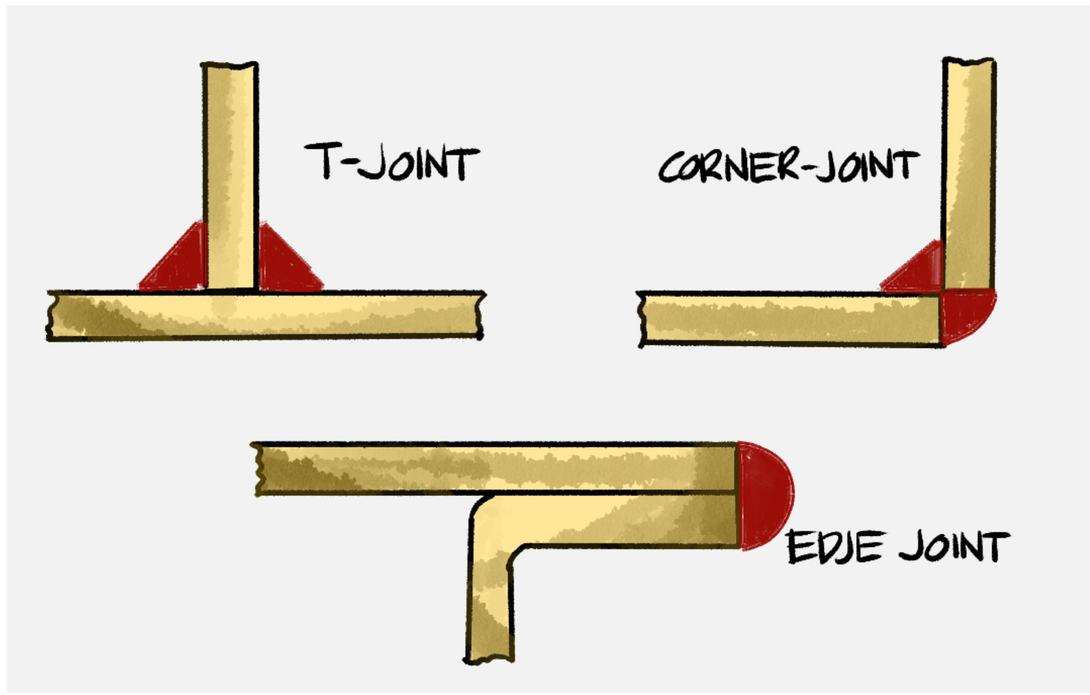
PARALLEL FILLET



BUTT JOINT



OTHER TYPES OF WELDS



STRENGTH OF BUTT WELDS

$F = S_t(t \times L)$

S_t = TENSILE σ
t = PLATE THCK.
L = LENGTH OF WELD

STRENGTH OF FILLET WELDS

$F = (S_s)(2L \times b\cos(45°))$

S_s = SHEAR σ
b = LEG DIM. OF WELD
$b\cos 45°$ = THROAT THCK.

FACTOR OF SAFETY $= \frac{S_{sy}}{S_s} = \frac{0.5S_y}{S_s}$

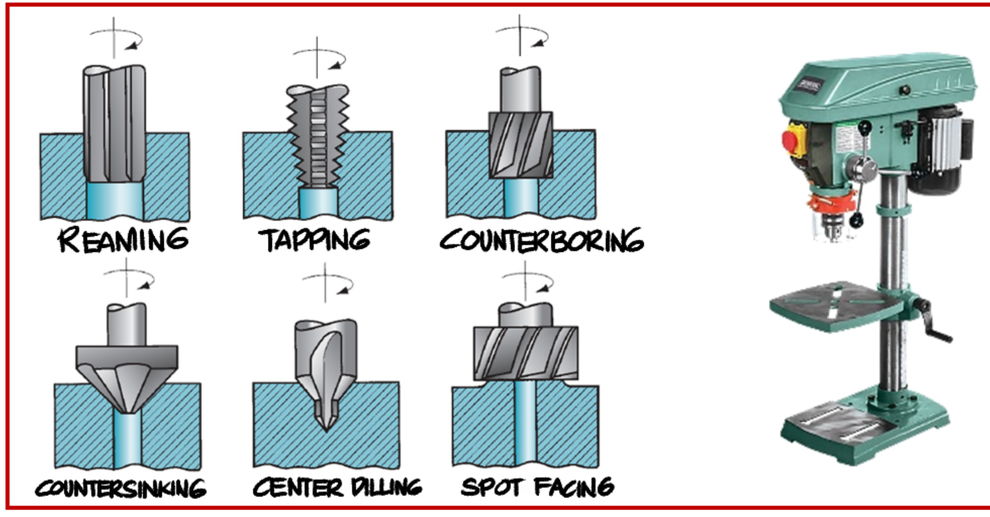
MAX. SHEAR STRESS $= 1.118S_s$

MAX. TENSILE STRESS $= 1.618S_t$

S_{sy} = SHEARING σ BASED ON YIELD
S_y = YIELD STRESS

MACHINE SHOP PRACTICE

DRILL PRESS STANDARD OPERATIONS



DRILLING — PRODUCING A HOLE BY REMOVING METAL FROM A SOLID MASS USING A CUTTING TOOL CALLED TWIST DRILL

COUNTERSINKING — PRODUCING A TAPERED OR CONE-SHAPED ENLARGEMENT TO THE END OF A HOLE

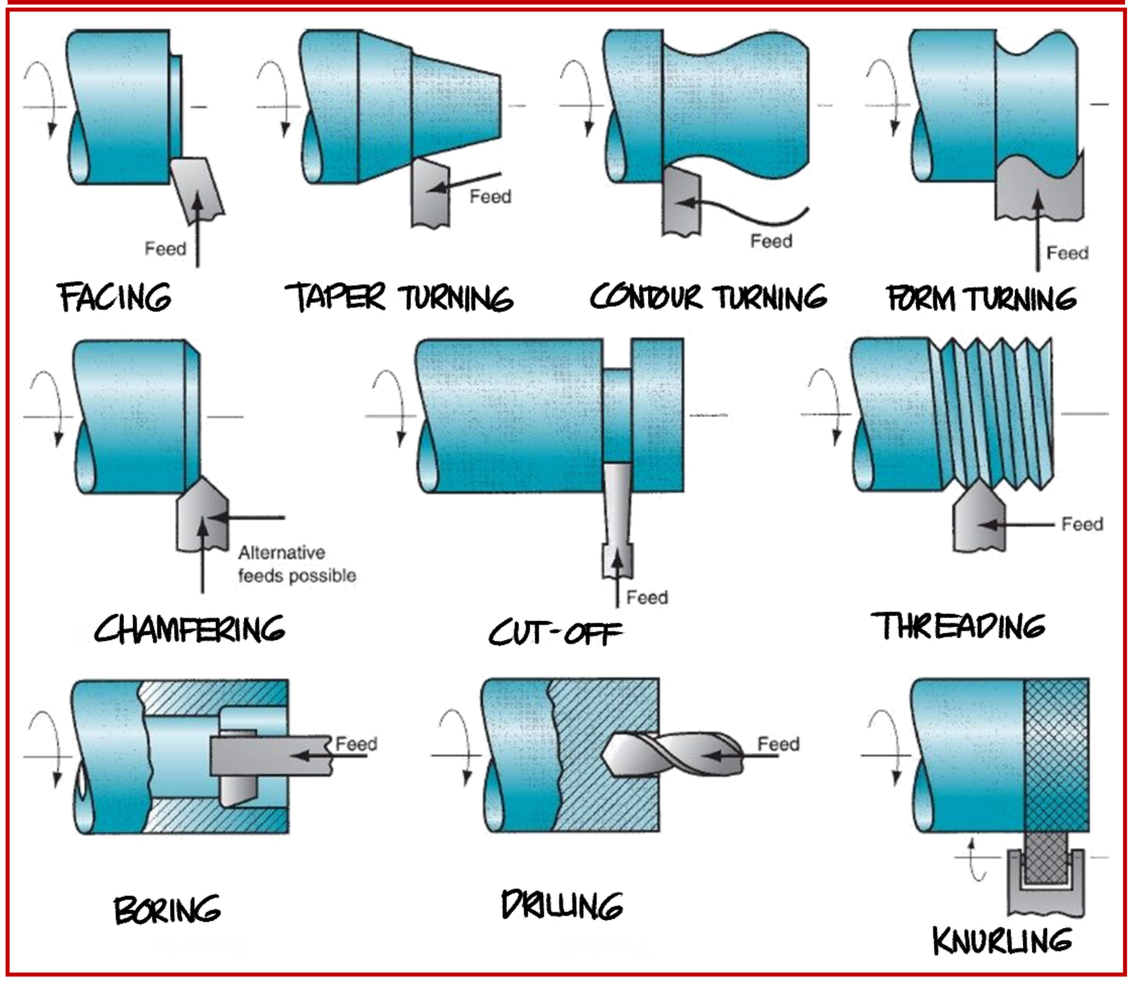
REAMING — SIZING AND PRODUCING A SMOOTH ROUND HOLE FROM A PREVIOUSLY DRILLED

SPOT-FACING — SMOOTHING AND SQUARING THE SURFACE AROUND A HOLE TO PROVIDE A SEAT FOR THE HEAD OF A CAP SCREW OR A NUT

TAPPING — CUTTING INTERNAL THREADS IN A HOLE WITH A CUTTING TOOL CALLED A TAP

COUNTERBORING — ENLARGING THE TOP OF A PREVIOUSLY DRILLED HOLE TO A GIVEN DEPTH TO PROVIDE A SQUARE SHOULDER FOR THE HEAD OF A BOLT OR A CAP SCREW

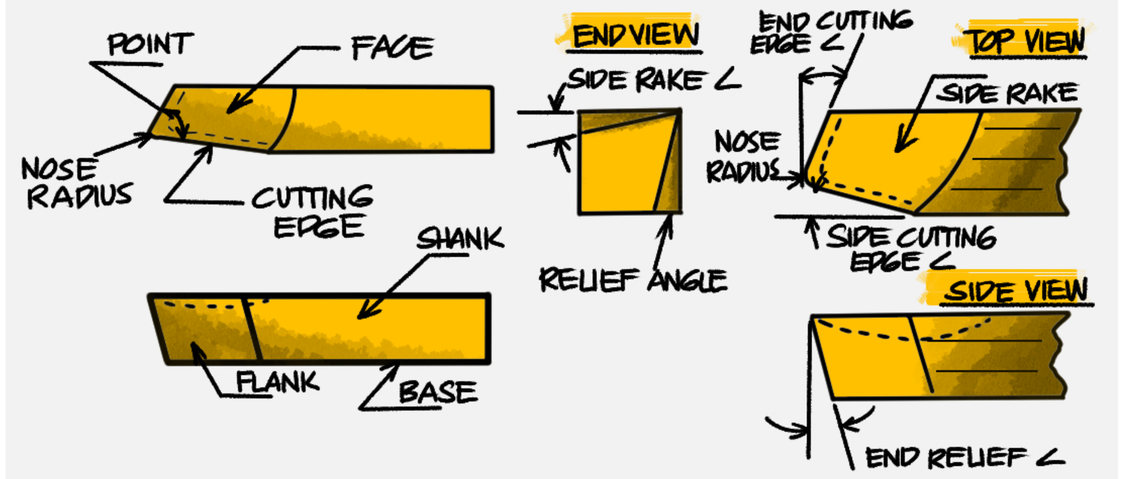
LATHE STANDARD OPERATIONS



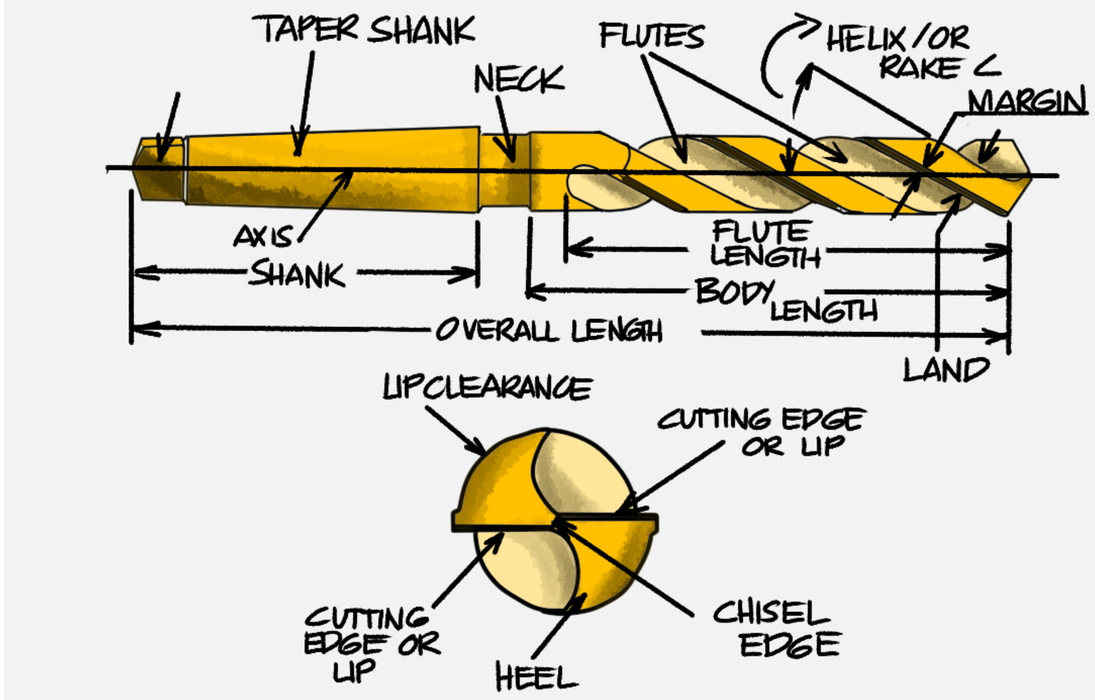
TAPER CALCULATIONS

$$\text{TAPER} = \frac{\text{LARGE } \varnothing - \text{SMALL } \varnothing}{\text{TAPER LENGTH}}$$

CUTTING TOOL



TWIST DRILL PARTS



CUTTING SPEED

$$V = \pi D N$$

D = DIAMETER, ft
N = SPEED, rpm

CUTTING TIME FOR TURNING, BORING, AND FACING

$$T = L / fN$$

f = FEED RATE, in/rev
N = LATHE SPINDLE SPEED, rpm

MILLING CUTTERS

$$T = \frac{6.3D}{W} = \frac{12.6D\cos A}{D + 4d}$$

D = CUTTER ø, in.
W = WIDTH OF CUT, in.
T = NUMBER OF TEETH
A = HELIX ∠ OF CUTTER
d = DEPTH OF CUT, in

MILLING MACHINE FEED RATE

$$f_m = f_t \times n_t \times N$$

f_m = MILLING MACH. FEED RATE, in/min
f_t = FEED RATE, in/TOOTH
n_t = NUMBER OF TEETH
N = SPEED, rpm

PLANNING TIME

$$T = \frac{W}{L}\left[L \times \left(\frac{1}{V_c} + \frac{1}{V_r}\right) + 0.025\right]$$

T = TIME, min
W = WIDTH OF SURFACE, in.
L = LENGTH OF STROKE, ft
V_c = CUTTING SPEED, rp
V_r = RETURN